GOVERNORS

AND

GOVERNING MECHANISM.

BY

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PREFACE.

THE first few chapters of this book give a general description of the principles underlying the action of the steam engine governor.

In discussing these, the ideal of "regularity" is substituted for that of isochronism. Also, difference of speed, and consequently *difference* of centrifugal force, is taken to be that on which the power of a governor depends.

Chapters IX. to XIV. deal with valve gears, and give a method of designing the governor in connection therewith.

The remaining chapters deal with governors in which the principles involved differ somewhat from those which underlie the action of the ordinary governor.

Two appendices have been added, one dealing further with governor power, and the other, suggested by the Editor of *The Practical Engineer*, giving numerous examples of governors and governing mechanism by well-known makers.

The writer wishes to express his best thanks to the Editor of *The Practical Engineer* for obtaining, and the various firms and gentlemen connected therewith whose governors and gear are described in the book for supplying, blocks and drawings for illustration, descriptive matter, and other information.

H. R. HALL.

Birmingham,

July, 1903.

ERRATUM.

Page 3, bottom line, for "centrifugal force," read "energy from which centrifugal force is derived."

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GOVERNORS

AND

GOVERNING MECHANISM.

Introduction.

No one ever thinks of writing on the subject of Governors without commencing with a description of the conical or revolving pendulum.

1. Therefore it will be necessary to commence this book with the time-honoured reference to the conical pendulum, and the use made of it by James Watt in setting it to operate the throttle valve of a steam engine, showing how he thus brought into mechanical science a very beautiful application of the law of supply and demand.

2. It is proposed to describe, according to custom, the astatic or isochronous "governor," and to attempt to show, contrary to custom, that this instrument is not a governor, because it does not possess one of the essential qualities of a governor; that, in fact, to say that a governor should be isochronous, or nearly so, is misleading.

3. It will be endeavoured to show what are the essential qualities that a governor should possess, and how these qualities affect the determination of the size and *power* of a governor for the *work* it has to do. In doing which it will be necessary to consider some of the various forms of valve gear with which the governor is connected, and which, together with the governor, form the governing mechanism.

4. It is intended to show how these qualities affect certain well known special forms of governor, such as the shaft governor, the inertia governor, gas engine governors, and relay governors.

CHAPTER I.

THE CONICAL PENDULUM.

IF two equal-length pendulums be set swinging through equal arcs at right angles to each other, in such a way that one comes to rest when the other is in the middle of its swing, and these two motions be combined, the result will be what is known as uniform circular motion.

Because: "Uniform circular motion is compounded of two simple harmonic motions of equal period and amplitude, taking place at right angles to each other and differing in phase by one quarter of the whole period." (Goodeve, "Elements of Mechanism.") And the motion of a pendulum is a simple harmonic motion. If, instead of two pendulums one and the same pendulum be set swinging sideways as well as forwards in the manner indicated, that is, if it is revolved it will, nevertheless, perform its vibration approximately in the same time as if left to swing forwards only.

The revolving or conical pendulum will describe a half-circle approximately in the same time that the swinging pendulum will make one vibration. (See note at end of Chapter II.)

From this fact and the law which gives the time that a pendulum of given length will take to make one vibration, the formula for finding the height of the cone or the point of suspension above the plane in which the balls revolve, of the conical pendulum revolving at a given speed may be derived.

This height is inversely proportional to the square of the speed, and if it is remembered that the height for sixty revolutions per minute is 9·78 in., or roughly, $9\frac{3}{4}$ in. (which is the length of a pendulum to vibrate twice per second,) it is only a matter of simple proportion to find the height for any other speed.

CHAPTER II.

Centrifugal Force.

To demonstrate this principle of the conical pendulum it is necessary to first consider the force which compels a body to move in a circle. Newton established "that whenever a body describes a circle with uniform velocity it must be subject to a constant force tending towards the centre of the circle."

The force which resists this is called centrifugal force. Centrifugal force is due to inertia, and the tendency of a moving body to continue its motion in a straight line and with unvarying velocity.

The attempt to change the motion of a mass is met by a resistance on the part of the mass which is exactly proportionate in amount to the rate of change, and is felt in a direction exactly opposite to that of the change of motion, that is, in this case, at right angles to the circumference of the circle. This force must not be confused with that derived from the energy possessed by a body moving with uniform velocity or with the energy stored up in a flywheel or other revolving body. That energy is the weight of the moving body multiplied by a certain height, *i.e.*, the height to which a body must be raised so that by falling from that height in vacuo it may acquire the velocity of the moving body.

The force which this energetic body may display depends on the rate at which it is stopped. Centrifugal force is due to a different cause. This is constantly acting and resisting the continual effort necessary to change the *direction* of motion of the mass from that of the straight line into that of the circle. It is a constant uniform bursting pressure, and acts in exactly the same way as pressure admitted to the inside of a cylinder.

Then the weight multiplied by the *height due to the rate at which a revolving body is constantly being pulled towards the centre of the circle* in which it revolves is the centrifugal force.

To find this in terms of the radius will necessitate a reference to proposition 36 of the third book of "Euclid," and sundry other matters: "If a line be drawn at a tangent to a circle, and from a point in this line outside the circle, another line be drawn through the centre of the circle, the rectangle contained by the whole line passing through the centre of the circle and the part of it outside the circle will be equal to the square on the part of the line drawn at a tangent to the circle."—"Euclid," prop. 36, book 3.

The distance that a body would be moved through by the force which compels it to move in a circle, in the time that it took to move through the part of the line drawn at a tangent to the circle, would be represented by that part of the line passing through the circle which lies outside it.

Let the distance = x, and the other distance, viz., that along the tangent = y, then

$$x\,(2\,r + x) = y^2, \text{ or } 2\,r\,x + x^2 = y^2.$$

Let the distance y be very small; y may be taken as equal to the arc of the circle, and x^2 may be neglected, therefore

$$2\,r\,x = y^2.$$

But the force which compels a body to move with uniform velocity is

$$\frac{w \times \text{velocity generated in one sec.}}{g};$$

or, putting f for the velocity generated in one second in the body of weight w by the action of the force F,

$$\mathrm{F} = \frac{w\,f}{g};$$

And the distance moved through by a body in a given time is

$$\frac{\text{time}^2 \times \text{vel. generated in one sec.}}{2},$$

therefore $$x = \tfrac{1}{2}\,\mathrm{ft.}^2.$$

Also, the motion in the circle is uniform, therefore

$$y = t \times \text{velocity}.$$

Therefore $2\,r \times \frac{1}{2}\,\text{ft.}^2 = t^2\,v^2$

and $$f\,r = v^2\text{—that is, } f = \frac{v^2}{r}.$$

Therefore $$\text{F} = \frac{w\,v^2}{g\,r}, \text{ or } w\,r\,\text{N}^2\ {\cdot}00034,$$

when w = weight,

v = velocity feet per second,

g = velocity feet per second acquired by a body in falling from rest in one second, viz., 32·2,

r = radius of circle in feet,

N = revolutions per minute.

So much for centrifugal force, which has been described at some length, because it not only enters into this demonstration of the principle of the conical pendulum, but is the *force on which almost all governors depend for their action,* to say nothing of its importance in connection with other mechanical matters.

It is now possible to proceed with the demonstration of the principle of the conical pendulum. Let D, fig. 1, represent the ball suspended at an angle by the arm C. D, when the governor is revolved, is acted on and held in equilibrium by three forces, two of which are known, viz., the weight acting downwards and the centrifugal force acting outwards; the third force is the pull of the arm C, and is the resultant of the other two.

By the principle of the parallelogram of forces, the radius r, or the distance of the ball from the centre of the point of suspension, bears the same proportion to the centrifugal force that the weight bears to the height h, or the point of suspension above the plane in which the ball revolves—that is,

$$\frac{w\,v^2}{g\,r} : w :: r : h;$$

i.e., multiplying together extremes and means, and eliminating the common factor w,

$$\frac{v^2 h}{g r} = r, \text{ or } v^2 h = g r^2;$$

this, again, is the same thing as

$$\frac{r}{v} = \sqrt{\frac{h}{g}},$$

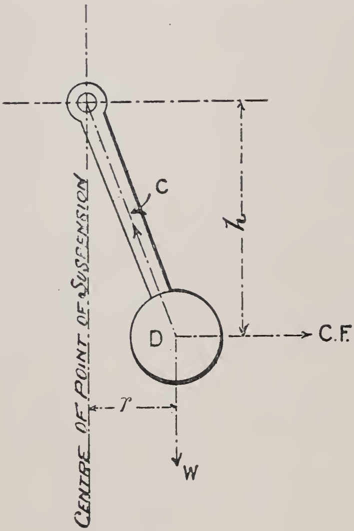


FIG. 1.

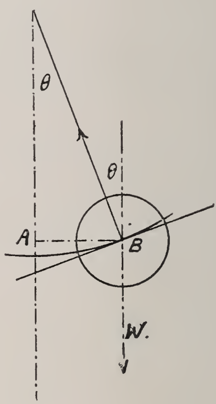


FIG. 1a.

and, as the motion is uniform, the time of a revolution multiplied by the velocity, or $t v$ is equal to the diameter of the circle, or $2 r \times 3{\cdot}1416$—that is to say

$$t = \frac{2 r \times 3{\cdot}1416}{v} = \frac{6{\cdot}2832 r}{v} = 6{\cdot}2832 \sqrt{\frac{h}{g}}.$$

Thus it is seen that the time of a revolution varies directly as the square root of the height of the cone, which is the same thing as saying that the height is inversely proportional to the square of the speed, as before stated.

Note on the time of swing of a pendulum communicated by Mr. W. Hewson, A.R.C.S. :—

The time of swing of a governor ball is

$$T = 2\pi\sqrt{\frac{h}{g}}$$

strictly—*i.e.*, a ball at a point with weightless arms, etc. But the time of swing of a pendulum is not exactly

$$T = 2\pi\sqrt{\frac{l}{g}}$$

not even for a simple ball suspended by a weightless string from a frictionless point of support, etc. The time varies with the angle, and is greater the greater the angle. The reasons being—

(*a*) That in the governor, the force on the ball holding it in towards the centre of the circle in which it swings is proportional to r, fig. 1, its distance from that centre, and therefore produces harmonic motion.

The balls moving in a circle, the points on a diameter of the circle which are projections of the balls on that diameter, move backwards and forwards as in harmonic motion.

(*b*) In a pendulum, on the other hand, the force is not proportional to the distance along the path to the position of equilibrium. See fig. 1a.

$$\text{In harmonic motion } T = 2\pi\sqrt{\frac{\text{distance}}{\text{acceleration}}}$$

$$\text{But in pendulum } T = 2\pi\sqrt{\frac{\text{length of pendulum} \times \theta \text{ (or arc)}}{g \sin\theta}}$$

$$\text{For acceleration} = \frac{\text{force}}{\text{mass}} = g \sin\theta \text{ along tangent.}$$

Now if arc ($l\,\theta$) were equal to A B ($l \sin\theta$) T would be equal to

$$2\pi\sqrt{\frac{l}{g}}$$

but it is not.

The nearer the arc approaches to A B, or the smaller the angle θ, the nearer is

$$T = 2\pi\sqrt{\frac{l}{g}}.$$

So the formula usually given for a simple pendulum is only true for indefinitely small arcs, and the true formula for a simple pendulum is not the same as for a governor except approximately.

CHAPTER III.

Watt's Governor.

The conical pendulum attached to a throttle valve is the oldest and most venerable form of centrifugal governor. As used by Watt, it consisted of two heavy balls carried at the ends of rods suspended from a central spindle, which was driven from the engine shaft by cord or belt. The arms were prolonged at the top, and connected by links to a collar which actuated a lever connected with a butterfly valve in the steam pipe, fig. 2.

When the balls rose, due to an increase of speed, the valve closed, diminishing the supply of steam so as to bring down the speed to its proper level. When the balls fell, owing to a decrease of speed, the valve opened, increasing the supply of steam so as to bring the speed up again. The principle of this action is that of supply and demand.

It is a curious fact that, while political economists recognise that for the proper action of the law of supply and demand there must be fluctuations, it has not generally been recognised by mechanicians in this matter of the steam engine governor.

The aim of the mechanical economist, as it is that of the political economist, should be not to do away with these fluctuations altogether (for then he does away with the principle of self-regulation), but to diminish them as much as possible, still leaving them large enough to have sufficient regulating power.

This, then, was Watt's governor, operating on the principle of supply and demand.

The principle on which the Watt governor operated is the same as that on which all simple governors act. By simple governors is meant those which render an engine in itself self-regulating. Auxiliary or supplementary governors, involving the principle of interference by some outside agency are generally too complicated to stand much chance of competing successfully with governors operating on this simple principle.

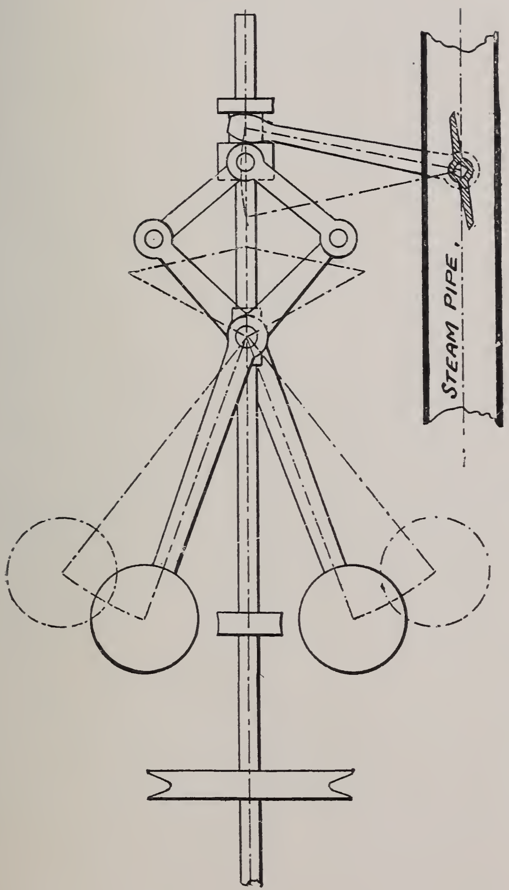


FIG. 2.

CHAPTER IV.

The Isochronous "Governor."

It will now be necessary to describe the Astatic or Isochronous "Governor."

A governor is said to be static when it will maintain the balls in a definite position for any given speed, and when it is capable of resisting external forces tending to alter their position. The *instrument* is *as*tatic when the balls assume indifferently any position at the same speed, and offer no resistance to external forces tending to alter their position. This is what is called the "Isochronous Governor," from the Greek "isos"—equal, and "chronos"—time. The isochronous governor may be called a one-speed, *any*-position governor, in contradistinction to the static, or one-speed *one*-position governor. Considering the uselessness of such an instrument for the purpose of a governor, it would be remarkable that so much time and thought should have been devoted by mechanicians to its construction, if it were not that it is a very beautiful piece of mechanism.

The simplest form of this instrument is the so-called parabolic *governor*. The parabolic governor was made in Germany (strictly speaking, Vienna), and brought over to this country in 1851. It is a modification of the conical pendulum. If, instead of allowing the balls of the pendulum governor to describe, as they rise, an arc of a circle, they are constrained to move in a parabolic curve such a governor would pass through its entire range at the same number of revolutions per minute. For it is a property of the parabolic curve that the height of the cone is always constant; and it has been shown that there is only *one* speed for one height of cone. If, therefore, in the governor opening out the height of the cone does not vary, the speed does not vary.

The commonest method of constructing an approximately parabolic governor is to cross the arms in such a way that their point of intersection rises by the same amount as the

plane of revolution of the balls. This is what is known as the X-armed type. (Fig. 3.)

A curious property of the X-armed governor is that it can be so constructed that the balls, instead of rising when the speed increases, will actually fall and describe a smaller

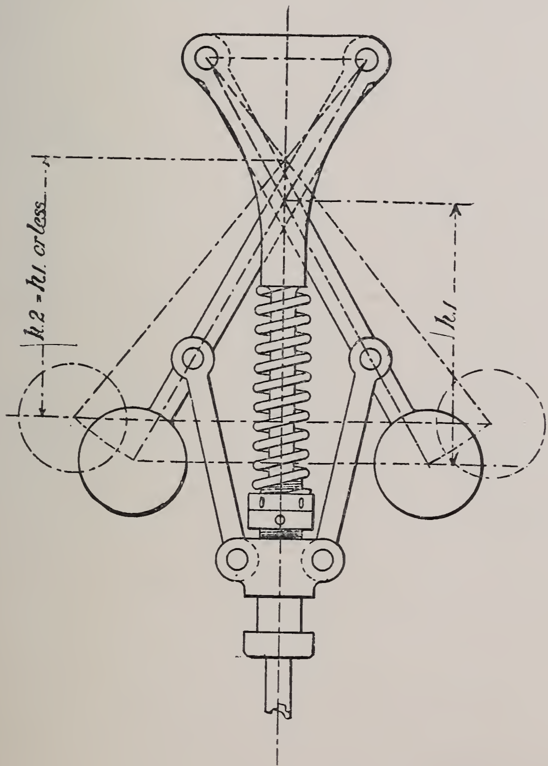


Fig. 3.

circle when running at a higher, than when running at a lower speed. It is merely necessary to arrange the points of suspension so that the arms cross in such a manner that the height of the cone *diminishes* as the balls descend; *i.e.*, to

make the distance between the points of suspension greater than the distance between the centres of the balls, as in fig. 4, and the rest follows from what has been seen as to the relationship between the height of the cone and the speed.

Another very beautiful example of this type, viz., the parabolic type, is the cosine governor, so called because the

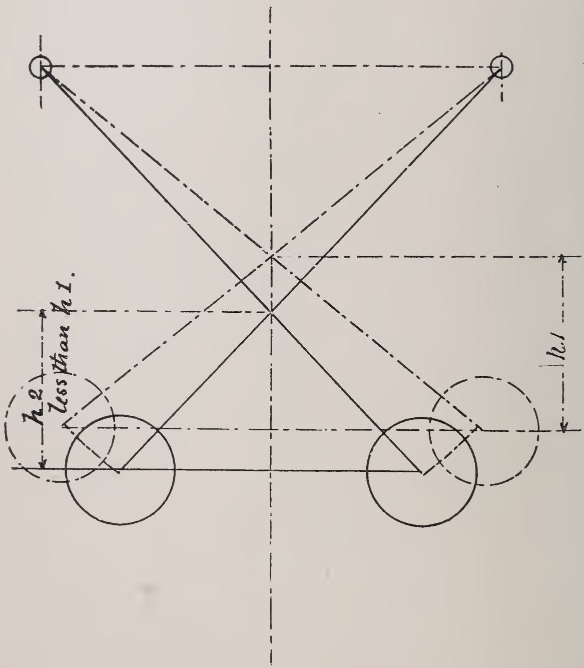


Fig. 4.

actual length of the arm must be multiplied by the cosine of the angle made by the hypotenuse to give the virtual arm. (Fig. 5.) The arms of this governor are two bell cranks, the ends of the short arms overlapping each other in such a way that lines drawn from them to the centre of balls will cross at a point below the plane of suspension. It

is possible to arrange the arms so that the point of intersection of these imaginary lines will rise by the same amount as the plane of revolution of the balls. This imaginary line is called the hypotenuse, hence this is sometimes called the hypotenuse governor.

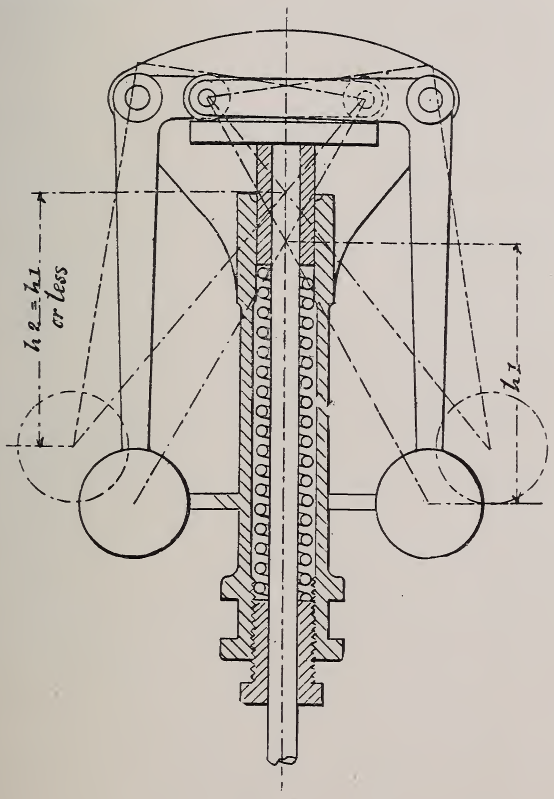


FIG. 5.

It is possible to arrange this governor so that it is in perfect isochronism for any length of range, whereas, in the X-armed type, the range is limited.

The isochronous governor, when running at its proper speed, offers no resistance to external forces tending to alter

the position of the balls, therefore it is not capable of maintaining the throttle valve or the valve gear in any one position, but will maintain them indifferently in any position, so accommodating is this governor (throttle valve wide open or quite closed), and on the slightest provocation or change of speed will run at its extreme top or bottom position, according to whichever way the speed may be changed. To remedy the unsteadiness of such a governor, as it is called, it is usual to provide it with a dash-pot.

The effect of the dash-pot is to prevent any immediate alteration in position of the balls when acted on by external forces, or by a change of speed; but it does not prevent their ultimate alteration to the extreme top or bottom position, and therefore does not make the governor any more capable of adjusting the steam supply arrangements to the requirements of the engine. The governor has still no mind of its own, so to speak. The dash-pot simply delays the movement of the governor, and delays are sometimes dangerous, especially when an engine is running away and it devolves on the governor to check it.

It was said that the principle of the action of all simple governors is that of supply and demand. When the greater demand made on the engine for power, which occasioned the fall of speed and, consequently, the greater opening of throttle valve is maintained, the fall of speed must also be maintained and *vice-versâ*. Therefore it will be necessary for the balls of a governor operating on this principle to occupy a definite position and maintain the throttle valve to a definite opening for a given speed, and it must be capable of resisting external forces tending to alter this position. Therefore, the governor must have one speed for one position, another for another. The difference of speed between any two positions need be only very small, in fact, the smaller the better; but a difference there must be. That is, the governor must be decidedly static. But the governor just described is isochronous or astatic, which is the opposite of what is required.

This leads to a consideration of the first and most important qualification that a governor should possess, and that is power. It is usual to say that a good governor must

be, first, "isochronous or nearly so." To say that a governor must be isochronous or nearly so, is equivalent to saying that it must have no power, or nearly none. It must have power to resist external forces tending to alter the position of the balls, and power to adjust the steam supply arrangements to the requirements of the engine.

CHAPTER V.

Qualifications that a Governor should Possess.

The qualifications that a governor should possess, then, are :

1st. Power, and to possess this, it must be static, or be a one-speed, one-position governor, and not a one-speed *any*-position governor.

2nd. Sensitiveness, *i.e.*, the difference in speed between the two extreme positions should be small.

3rd. Regularity, *i e.*, the differences in speeds between any two equal intermediate positions should be *alike.*

4th. Steadiness, and

5th. Lightness, which, though not an essential quality, is still a very desirable one.

It will be endeavoured to show that the sensitiveness of a governor depends on the degree in which the first and third of these properties, viz., power and regularity, are possessed by a governor, and *not* on its approximation to isochronism as is generally supposed.

CHAPTER VI.

Power.

The power available in any ordinary centrifugal governor to do the work of shifting the eccentric or valve, to give a greater or less admission of steam, for a heavier or lighter load, is the mean of the difference of the centrifugal force of

the balls at two different speeds of the governor or engine, within which it is allowed to run, multiplied by the distance between the two positions of balls.

If two governors be taken for comparison, each having the same percentage of variation, that is, the same sensitiveness and the same range, then the power of the one is to that of the other as the *difference* of centrifugal force in each case. And these differences are, it can be proved, one to the other as the mean centrifugal force in each case.

That is, if $a : b : c : d$,

$$\text{then } a^2 - b^2 : c^2 - d^2 :: \left(\frac{a + b}{2}\right)^2 : \left(\frac{c + d}{2}\right)^2$$

for by componendo and dividendo:

$$\frac{a - b}{a + b} = \frac{c - d.}{c + d.}.$$

Therefore :

$$\frac{(a + b)\ (a - b)}{(a + b)^2} = \frac{(c + d)\ (c - d)}{(c + d)^2}.$$

Therefore

$$\frac{(a + b)\ (a - b)}{(c + d)\ (c - d)} = \frac{(a + b)^2}{(c + d)^2} = \frac{\dfrac{(a + b)^2}{2}}{\dfrac{(c + d)^2}{2}}.$$

Therefore

$$\frac{a^2 - b^2}{c^2 - d^2} = \frac{\left(\dfrac{a + b}{2}\right)^2}{\left(\dfrac{c + d}{2}\right)^2}$$

Then the power of the one governor is to that of the other as the centrifugal force simply, taking the mean speed in each case; that is, the relative power of any two governors of the same sensitiveness, and with balls describing the same radii, can be readily compared by taking in each case the weight of the balls and multiplying by the square of the speed.

A governor with balls of twice the weight of another, running at the same speed, will have twice the power, but a governor with balls of the same weight, running at twice the speed, will have four times the power. The readiest way, then, to increase the power of a governor, is to increase the speed.

It will now be seen why the conical pendulum is not often used in these days of high speeds and high pressures, as a governor. Not because it is not isochronous, as is sometimes said, but because of the difficulty of increasing its power without enormously increasing the weight. Let it be required to run the Watt governor at 200 revolutions per minute—a not uncommon speed for a governor—the height of the cone would be ·88 in. This is obviously impracticable.

A further point relating to the Watt governor may be here mentioned. It was seen that the height of the point of suspension above the plane in which the balls revolve was inversely proportional to the squares of the speeds, and it is now said that the power of a governor, the weight being the same, is in proportion to the square of the speed; therefore the height of the cone in the Watt governor is inversely proportional to the *power*.

The power of a governor depends on the centrifugal force, but if the ball is to be maintained in any one position it must be in equilibrium in this position; that is, it must be acted on by another force which exactly balances this. In the pendulum governor this was the weight of the ball itself.

If it is required to increase the centrifugal force without increasing the weight of the ball, then some other force must be introduced; that is, the governor must be loaded. All modern governors, of whatsoever so-called variety, are of this class. The load may consist of a dead weight simply, or a spring simply, or a combination of the two.

In the Porter governor, fig. 6, it consists of a dead weight simply, hung on the central spindle, the arrangement of the arms being similar to the Watt or X-armed variety. In this case the weight of the balls themselves contribute something towards the resistance to centrifugal force; the

remainder—by far the greater portion—being made up by the dead weight or counterpoise, as it is called.

In the Pickering governor, fig. 7, the loading is not so apparent, but consists of the resistance to bending of the flat laminated springs which constitute the arms of this governor.

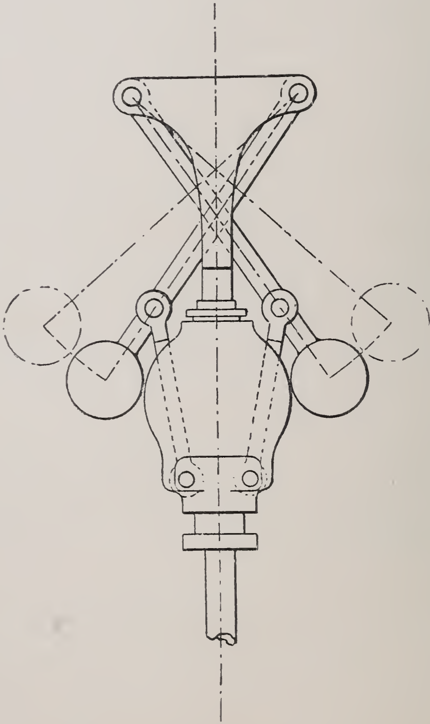


Fig. 6.—Porter Governor.

Fig. 8 is a spring loaded governor of the Hartnell type, and fig. 9 is a governor in which the load consists of a combination of spring and dead weight.

This governor is a modification of the bell crank lever type, with a rising head forming the dead weight part of the load-

ing, and consists of the head A carrying the bell crank levers and balls B; the spindle C driven by bevel wheels and

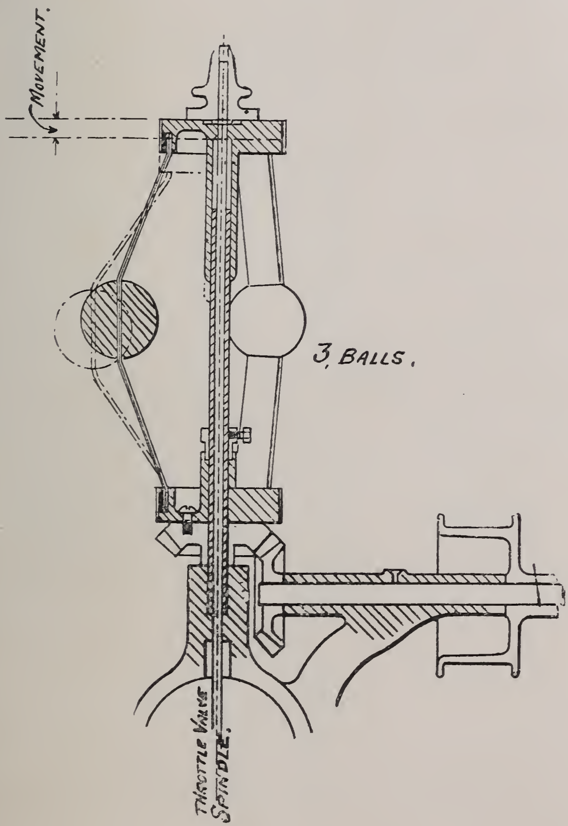


FIG. 7.—Pickering Governor.

pulleys from the crank shaft the fixed stem D, on which the head revolves, and the spring E.

A washer forming an oil box F, makes up the distance between the face of stem on which it turns, and the head

of the spindle driving the governor. This also forms a guide on which the bored part of the governor head slides.

The centrifugal force of the balls B, pressing on the top of the spindle through the bell crank levers, lifts the head

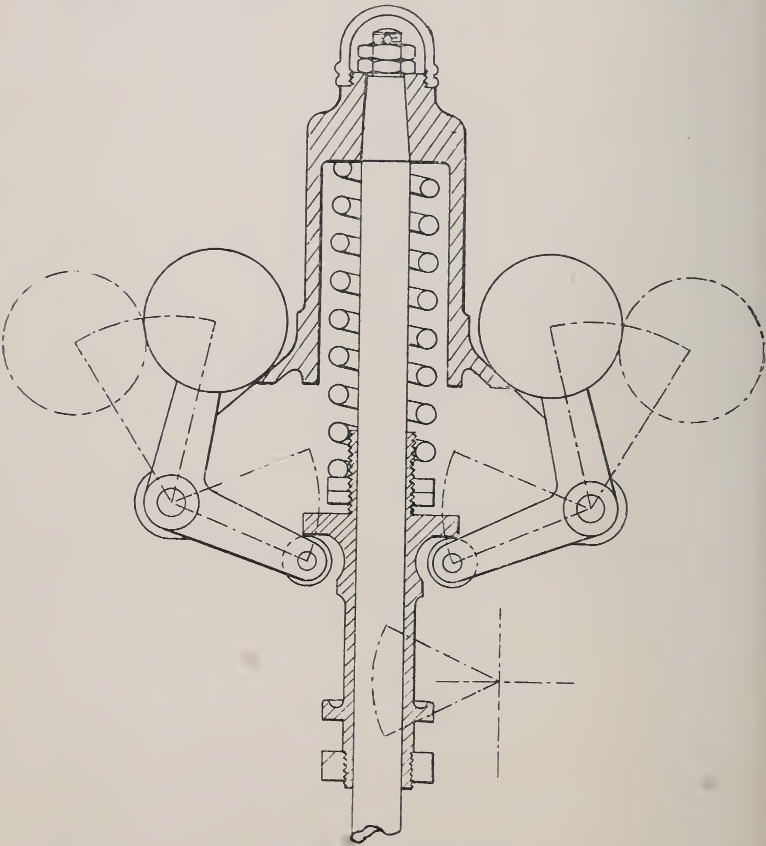


Fig. 8.—Hartnell Governor.

compressing the spring against the underside of the washer F. The lift of the head is given to the expansion valve through the levers and rod as shown in fig. 22, Chapter XI.

Loaded governors are the only governors with which any one, in these days, need concern himself, and the only thing

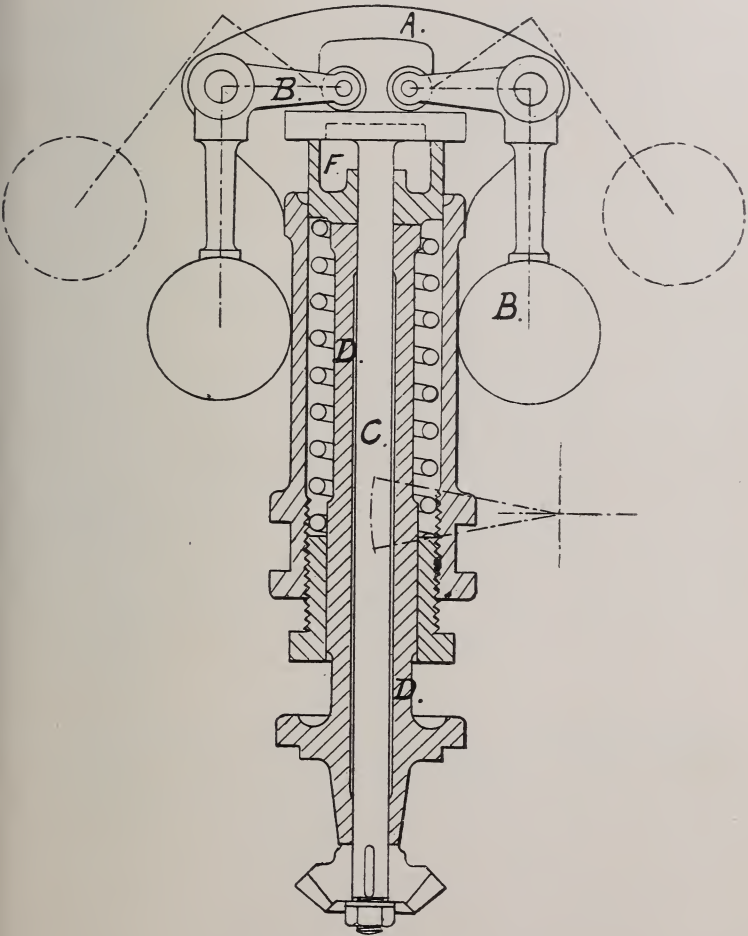


Fig. 9.

to remember concerning them is that the moment of the load must equal the moment of centrifugal force ; that is,

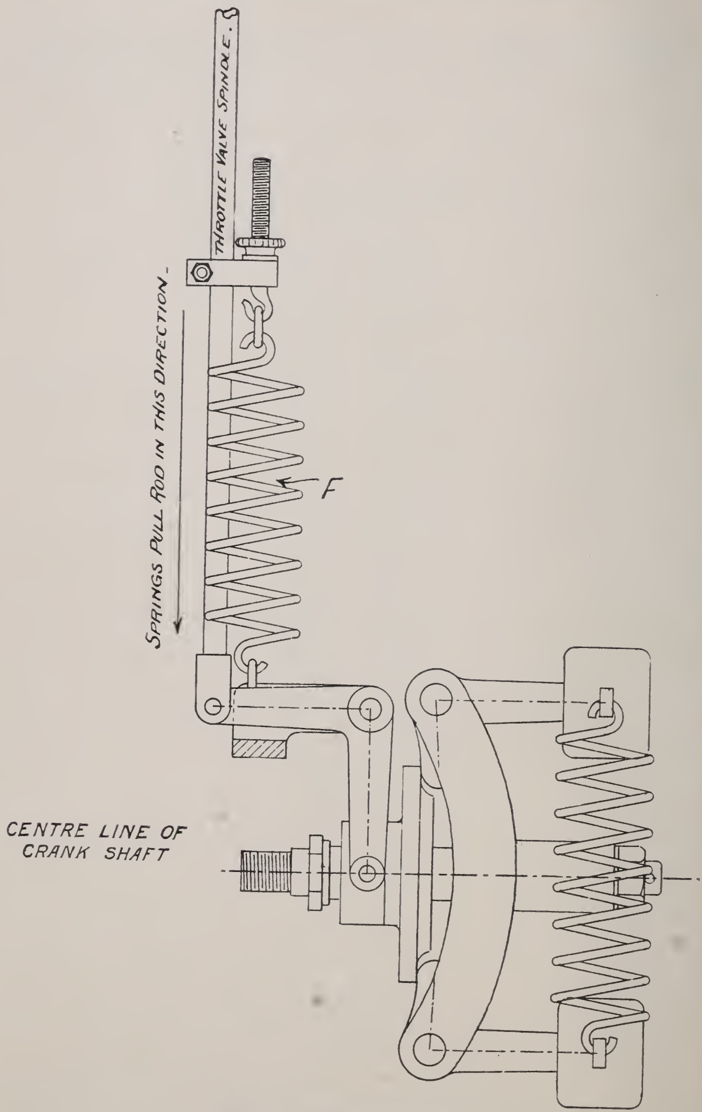


FIG. 10.—Willans Governor.

the load multiplied by the distance through which it is moved must equal the centrifugal force multiplied by the distance through which it acts.

In a governor arranged like Willan's, fig. 10, with balls attached to opposite ends of the same spring or springs, it is only necessary to consider the centrifugal force due to one ball, because the balls pull one against the other—the centrifugal force of one forming the reaction for the centrifugal force of the other.

The spring or springs would require to exert only half the force with twice the extension of a spring arranged in the ordinary way; that is, with the centrifugal force of both balls acting in the same direction, the spring supported at the other end against a fixed stop.

Since it is difficult to make a spring within the length available that will give sufficient extension with so small an increase of resistance, this governor is fitted with compensating springs F, on the throttle valve spindle, which act with the centrifugal force, causing the loading to increase at a more conveniently rapid rate. These springs being outside the governor also afford a means of varying the resistance, hence the speed of the engine while running.

It will now be seen that the power of governors, having the same percentage of variation, and the same range, may also be compared by comparing the *loads;* that is, the weight of the counterpoises, or the strength of the springs.

Sensitiveness.

In addition to a governor being powerful it should be sensitive; that is, the difference in speed between the two extreme positions taken up by the balls should be small. It was seen that the power of a governor depended on the *difference* of centrifugal force of the balls at two different speeds. If it is required to do more work with a governor, without increasing the mean speed, it can be done by increasing the difference in speed between the two extreme positions. But this is to increase its variation, or diminish its sensitiveness. If, therefore, it is required to increase its sensitiveness, still leaving it capable of doing the same work,

the *power* of the governor must be increased. The sensitiveness, then, depends upon the power.

A governor running at 320 revolutions mean speed, and capable of shifting 180 lb. through one inch with a 4 per cent variation, would require to run at 340 revolutions mean speed to be capable of doing the same work with a 2 per cent variation; that is, the power would require to be increased by about $12\frac{1}{2}$ per cent, or $\frac{1}{8}$th, to diminish the variation one half.

CHAPTER VII.

Regularity.

The next thing a governor should possess is regularity, *i.e.*, the difference in speeds between any two equal intermediate positions within the total range of variation should be alike.

This is a point to which very little attention has been paid by engineers, with the result that very few governors possess this property, and are, consequently, much less powerful with the same sensitiveness than they might be if properly designed.

The following are the actual variations in speed of a 9 in. by 12 in. single cylinder engine, boiler pressure 85 lb. per square inch, from a table given in Proc. Inst. Mechanical Engineers, page 230, No. 2, 1895, and may be taken as a fair example of what occurs in practice :—

Engine with Throttle Valve.

B.H.P. ......20·04, 17·82, 15·54, 13·43, 11·19, 8·9, 6·61, 0.
Difference...2·22, 2·28, 2·11, 2·24, 2·20, 2·19, 6·61.
Speed ......147·2, 147·7, 148·1, 149·9, 151·2, 152·2, 154·1, 155.
Difference...·5, ·4, 1·8, 1·3, 1·0, 1·9. 1·9.

Engine with Auto or Variable Cut-off.

B.H.P. ......20·5, 18·25, 16·04, 13·68, 11·34, 8·98, 6·60, 0.
Difference...2·25, 2·21, 2·36, 2·34, 2·36. 2·38, 6·60.
Speed ......150·5, 151·3, 152·6, 152·8, 153·3. 153·6, 153·8, 155·1.
Difference...·8, 1·3, ·2, ·5, ·3, ·2, 1·3.

In the first the total variation is 5·1 per cent, and the least variation ·4 revolutions. If this least variation were sufficient to govern on, between 17·82 and 15·54 B.H.P., it is only fair to assume that it would have been sufficient for the same variation of power in any of the other positions. If the governor had been regular with ·4 revolutions difference between each position, the total variation could have been reduced to 1·8 per cent. In the automatic engine the case is even worse. Fig. 11 shows the above variations

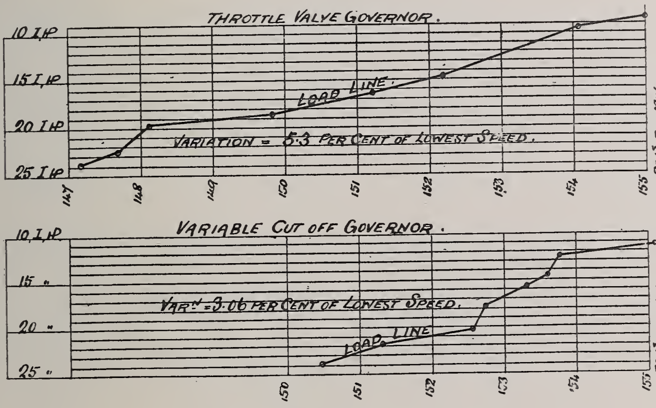


Fig. 11.

in speed, plotted as curves, and fig. 12 shows the same for a compound double-cylinder portable engine, with cylinders $7\frac{3}{4}$ in. and $11\frac{1}{2}$ in. diameter by 12 in. stroke, boiler pressure 135 lbs. per square inch, taken from the same source.

It has been seen that the total power of a governor depends on the difference of centrifugal force when running at its two extreme positions, corresponding to full load and no load in the engine; but in practice it is very seldom that the full load is thus suddenly thrown off or on, and that the governor is called upon to exert its full power. In regular work the load will be more likely to vary between quarter or three-eighths to five-eighths or three-quarters of the maximum, and the governor will only be called upon to regulate the supply of steam to suit these variations, that is,

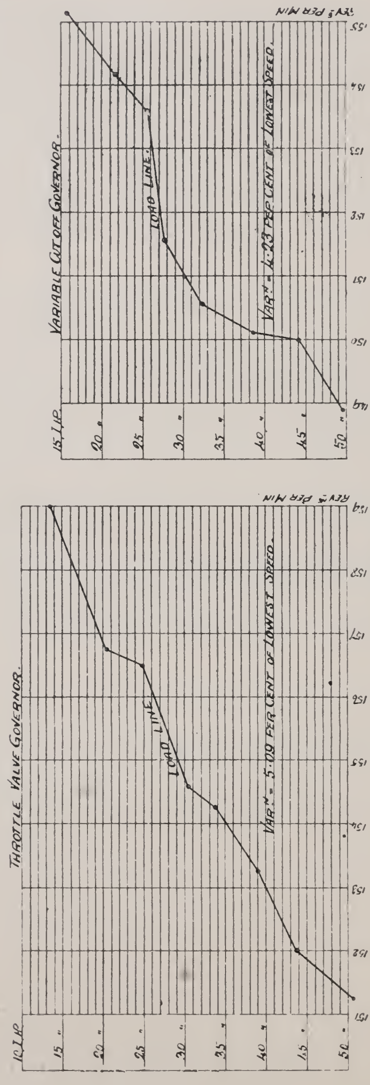

VARIABLE CUT OFF GOVERNOR.
LOAD LINE.
VARN = 4·23 PER CENT OF LOWEST SPEED.
REVS PER MIN
15 I.H.P.
20 "
25 "
30 "
35 "
40 "
45 "
50 "
THROTTLE VALVE GOVERNOR.
LOAD LINE.
VARN = 5·09 PER CENT OF LOWEST SPEED.
REVS PER MIN
10 I.H.P.
15 "
20 "
25 "
30 "
35 "
40 "
45 "
50 "

FIG. 12.

to take up intermediate positions corresponding to these fractional loads. The power of the governor to do this work depends on the difference of centrifugal force between these intermediate positions.

If now, the difference in speed between any two of these *middle* intermediate positions is less than it is between the *top and bottom* intermediate positions, as is nearly always the case in governors, then the power of the governor at these positions will be less than it is at the top and bottom positions, *and less than it would be if the differences were all alike.* That is, for the governor to have as much power at *any* intermediate position, the total variation must be increased by as much as the least difference of speed falls below the mean. If this difference amounted to one-fifth, as it often does, then the total variation would have to be three times as much.

Thus, then, a governor should be regular, and the sensitiveness depends on the *power* and the *degree of regularity* possessed by a governor. To construct a governor that shall be perfectly regular, it is merely necessary to see that the lifts necessary to make the equal alterations in load are arranged so that the governor will run with equal differences of speed between these positions. In other words, to arrange that the movements shall be proportional to the speeds.

Although it has not been generally recognised, one of the chief causes of irregularity in spring-loaded governors is the spring itself. A well-known writer on the steam engine says: "In any of this class of governors (viz., spring loaded) it is possible to secure absolute theoretical isochronism, because the centrifugal force of a body revolving in a circle varies directly as its distance from the centre for a given uniform speed, and the resistance of a coiled spring also increases directly as it is compressed." This possibility of isochronism in a governor seems to have covered a multitude of sins, for let it be shown that a governor could be made isochronous, and nothing more was supposed to be required than to make it just not so. Let it be admitted that a governor must be just *not* isochronous, and this is where the trouble comes in, because then the speed must increase, and the centrifugal force varies not only directly as the radius, but also unfortunately as the square of the speed.

Suppose the range of a governor to be 2 in.; let it be divided into four equal parts, and let it be required to run with one revolution difference between each of these five positions, viz., at 198, 199, 200, 201, and 202 revolutions per minute, corresponding to a total variation of 2 per cent. The differences of the squares of these numbers are 397, 399, 401, 403, or as 7, 9, 11, 13; that is, the spring would have to compress through the first ½ in. with an addition of, say, 7 lb., through the next ½ in. with an addition of 9 lb., through the next with 11 lb., and the next 13 lb.

But the resistance of a spring varies directly as it is compressed; that is, if it takes 7 lb. to compress it through the first ½ in., it will only take 7 lb. more and not 9 lb. to compress it through the next ½ in. and so on.

What is wanted, then, is a spring that will compress at a varying rate, so that the loads are not in a simple but in a sort of quadruple compound arithmetical progression. Thus the loads in the present case are 7, 16, 27, 40. An approximation to this might be obtained by using four springs, one offering a resistance at the rate of 7 lb. to the ½ in. compression or extension, and three each offering a resistance of 2 lb. to the ½ in. compression or extension, arranged to come into action in the following order:—

0 lb.	7 lb.	14 lb.	21 lb.	28 lb.	resistances of spring 1.
	0	2	4	6	,, ,, ,, 2.
		0	2	4	,, ,, ,, 3.
			0	2	,, ,, ,, 4.
—	—	—	—	—	
0 lb.	7 lb.	16 lb.	27 lb.	40 lb.	total resistances.
	½ in. ×	½ in. ×	½ in. ×	½ in.	

NOTE.—The actual resistances would, of course, be much greater than given here, but would vary in the same proportion.

As the construction of a spring of this description may present some difficulty, it will be better to see if the same result cannot be arrived at by a proper arrangement of the valve gear; and all that is necessary is to arrange that the lifts of the governor (necessarily unequal) for these equal differences in speed shall correspond with the several posi-

tions of the valve gear necessary for the proper supply of steam for equal differences of load. In other words, the valve gear must be made to suit the governor. It may be thought that such a simple and obvious condition for a governor would never be overlooked, but such is not the case. The writer knows of only two governors in which it is fulfilled. All that need be said further in regard to this is that it is an easier matter to arrange for this condition with auto-gear, or variable cut-off, than with a throttle valve.

CHAPTER VIII.

Steadiness.

One of the chief causes of unsteadiness in a governor is the unsteadiness of the engine itself. An engine may perform a certain number of revolutions per minute with unvarying regularity, and yet, at the same time, it may be turning a great deal faster during one part of each revolution than another; that is, it may run jerky. The flywheel is to prevent this. The heavier the flywheel the less the jerkiness, but it is impossible to do away with it altogether.

A good single-cylinder engine may have as little as 2 per cent jerkiness. Any less would mean an excessively heavy flywheel, but a good governor feels this more or less, depending on its sensitiveness.

A spring-loaded governor, loaded for a variation of 2 per cent, if perfectly free from friction, dash pots, &c., would be jerked through its whole range every time this variation took place. A governor loaded for 4 per cent would be jerked through half its range—for 8 per cent through a quarter, and so on. In practice, however, the vibration would not amount to as much as this, as it is impossible to eliminate friction, resistance of valves, &c.; but these factors are variable and uncertain, and cannot be relied upon to ensure steadiness in a governor, besides, they ought not to exist.

Supposing, now, part of the loading to consist of dead weight, then the movement of the governor, during any

momentary fluctuation of speed, would depend upon the space that the weight could be moved through when acted on by the force generated by the increase of speed in the time. But the time is of very short duration. If the speed of the engine is 150 revolutions per minute, and the full force due to the increase of speed be taken to act during half the time between the fluctuations, it will be acting for one-tenth part of a second.

If the dead weight in the governor and connections be only equal to the force due to the change of speed, then the movement will be equal to the space described by a weight when acted on by gravity, viz., its own weight in falling, for one-tenth of a second. But if the weight be, say, five times this amount, then the movement will be equal to the space described by a weight when acted on by one-fifth gravity, or one-fifth its own weight for one-tenth of a second. The movement would amount to about $\frac{7}{16}$ths of an inch, this is supposing the governor to be perfectly free.

Example of steadying effect of inertia of parts: Coefficient of steadiness due to weight of flywheel = 50. This, at a speed for engine of 150 revolutions per minute, equals a variation of 2 per cent 300 times a minute. The increase of centrifugal force, due to 2 per cent increase in speed with balls at the same radius (in the governor taken for this example, see fig. 9) was 5·624 lb., and this equalled a lifting force of 11·24 lb., acting five times a second for, say, one-tenth of a second.

The dead weight in the governor equals, say, 50 lb., and the space described by a weight of 50 lb. in one-tenth of a second, when acted on by a force of 11·24 lb.,

$$= \frac{1}{2} \times \left.\frac{11{\cdot}24}{50}\right\} gt^2 = {\cdot}0359 \text{ ft.} = \frac{7}{16} \text{ in.}$$

NOTE.—In this example the weight forms part of the loading of the governor, but the result as regards the time taken to move the mass is the same, whether the dead weight acts with or against the loading of the governor, or whether it is balanced in itself, as it might be if in a horizontal position forming part of the valve gear. It is better, however, to constitute this dead weight part of the

loading of the governor, as it then serves the additional purpose of diminishing the strength of spring required.

Thus the chief cause of unsteadiness in a governor is the unsteadiness of the engine itself, and to obviate this, part of the loading should consist of dead weight. The shaft governor is deficient in this respect. The shaft governor is a pure spring governor. Owing to its peculiar arrangement on the shaft of the engine it is difficult to load it otherwise. To obviate this deficiency the shaft governor is often constructed with a dash pot. But dead weight may be introduced forming, in this case, part of the eccentric or valve gear.

Lightness.

It has been seen that a deficiency of dead weight in the loading of a governor means a deficiency in steadiness; but, on the other hand, lightness is a thing to be desired in a governor. According to a recently issued spring governor maker's circular, a Porter governor, which is purely a dead-weight governor, would weigh seven and a half times as much as a pure spring governor of the same power. But this governor would be extremely steady.

It seems, then, that a sort of compromise should be made between extreme lightness and extreme steadiness in governors, so that a governor should be light, and yet have sufficient dead weight to prevent its being unduly affected by the jerkiness of the engine; in fact, a certain amount of unsteadiness may not be altogether a disadvantage, as it keeps the joints free so that the governor is ready to respond immediately any permanent alteration of speed takes place. What exactly this amount should be is best found from practice.

CHAPTER IX.

Governor Gear.

In order that a governor may perform the operation of regulating the supply of steam to an engine, it must be connected with some form or other of valve gear. Moreover, it must be adapted to, and designed in connection with, that gear.

Generally speaking, it may be said that a governor regulates the supply of steam to an engine in either of two ways, viz. :—(1) By operating a throttle or reducing valve, thus reducing the pressure of steam in the steam chest before it is admitted to the piston. (2) By varying the cut-off, thus admitting steam to the piston at full pressure, but in varying amounts. In the first case the governor is known as the throttle valve governor; in the second case it would be one or other of the various classes of "Automatic" governors.

Automatic governors may again be subdivided into two classes, depending on the class of steam-distributing gear to which they are attached, viz. : (*a*) Those used in connection with some form of slide-valve gear; and (*b*) those used in connection with trip gears.

CHAPTER X,

Throttle Valve Gear.

The action of the throttle valve is to reduce the pressure of steam in the steam chest according to the load on the engine, by partly closing the steam inlet, the cut-off remaining the same.

With a constant cut-off, say five-eighths of the stroke, and loads varying as 100, 75, 50, and 25, the mean pressure in the steam chest would require to be reduced by the

throttle valve in practically the same ratio as the loads, to give correspondingly mean effective pressures on the piston. The terminal pressures, however, at this cut-off, with these initial pressures, would be less than with cut-off gear, as the steam pressure at the point of cut-off would fall considerably below the *mean* initial pressure, depending on the size of the steam chest. With a small steam chest the terminal pressures, and consequently the steam consumption for the same mean effective pressures, would be less than with a

Diam. of steam pipe,	Diamater of governor balls.	Revolutions per minute.			Total variation.	Diameter of path of balls.	Lifting power of governor.
		Min.	Mean.	Max.			Ft-lbs.
Inches.	Inches.				Per cent.	Inches.	
$1\frac{1}{4}$	$1\frac{13}{16}$	438·75	450	461·25	5	$2\frac{11}{16}$ to $4\frac{7}{16}$	·529
$1\frac{1}{2}$	$2\frac{1}{8}$	409·5	420	430·5	5	$3\frac{5}{8}$ to $5\frac{5}{8}$	·98
$1\frac{3}{4}$	$2\frac{1}{8}$	409·5	420	430·5	5	$3\frac{5}{8}$ to $5\frac{5}{8}$	·98
2	$2\frac{1}{8}$	409·5	420	430·5	5	$3\frac{5}{8}$ to $5\frac{5}{8}$	·98
$2\frac{1}{2}$	$2\frac{3}{8}$	370·5	380	389·5	5	$4\frac{1}{8}$ to $7\frac{1}{8}$	2·47
3	$2\frac{3}{4}$	312	320	328	5	$4\frac{1}{2}$ to $7\frac{3}{4}$	3·25
$3\frac{1}{4}$	$2\frac{3}{4}$	312	320	328	5	$4\frac{1}{2}$ to $7\frac{3}{4}$	3·25
4	$3\frac{1}{8}$	312	320	328	5	$5\frac{3}{8}$ to $8\frac{7}{8}$	5·45

FIG. 13.

Table of powers of some "Pickering" type of governors with from 3 to 5 per cent variation, as made by Messrs. Tangye.

larger steam chest. In arranging, therefore, an engine for a throttle valve governor, the space between the piston and throttle valve should be kept as small as possible.

Supposing now the rate of flow of steam to be the same at all positions of the throttle valve, then the openings necessary for it to make, in order to maintain the above reduced pressures in the steam chest, would exactly correspond with the loads, viz. : at half-load the throttle valve would be half open, and so on. But the flow of steam is not the same at all positions of the throttle valve; it becomes

faster as the pressure in the steam chest is diminished, or, in other words, as the throttle valve closes. Therefore it is necessary either to make the throttle valve governor with movements smaller in its higher positions (throttle valve nearly closed) for the same alterations of speed, and larger in its lower positions (throttle valve nearly full open); or to so design the throttle valve that it will give a less opening for the same movement when nearly closed, than when nearly full open.

The power required to operate a throttle valve is much less than that required to operate some other kinds of valve gear. The valve itself is usually an equilibrium valve, and the work to be done consists mainly in overcoming the friction of the spindle and stuffing box, pins, &c.

The table, fig. 13, gives the power of Pickering governors for different sizes of throttle valve, with percentage of variation.

CHAPTER XI.

"Automatic" Slide Valve Gear.

A varying cut-off may be obtained:

1. By a single slide valve having variable travel and angular advance of eccentric but constant lap.

The varying travel and advance may be given:

(*a*) By a single eccentric shifting on the shaft and operated by a shaft governor. (Example, Fig. 14.)

(*b*) By two eccentrics fixed on the shaft but giving the variable travel and equivalent angular advance by means of a link and die, as in an ordinary link motion reversing gear. (Example, fig. 15.)

(*c*) By a form of radial gear as Joy's valve gear, the Allen link, &c. (Examples, fig. 16 and 16*a*.)

A varying cut-off may be obtained:

2. By two slide valves having variable relative travel but constant negative lap on expansion valve and constant angular advance of eccentrics, the variable relative travel

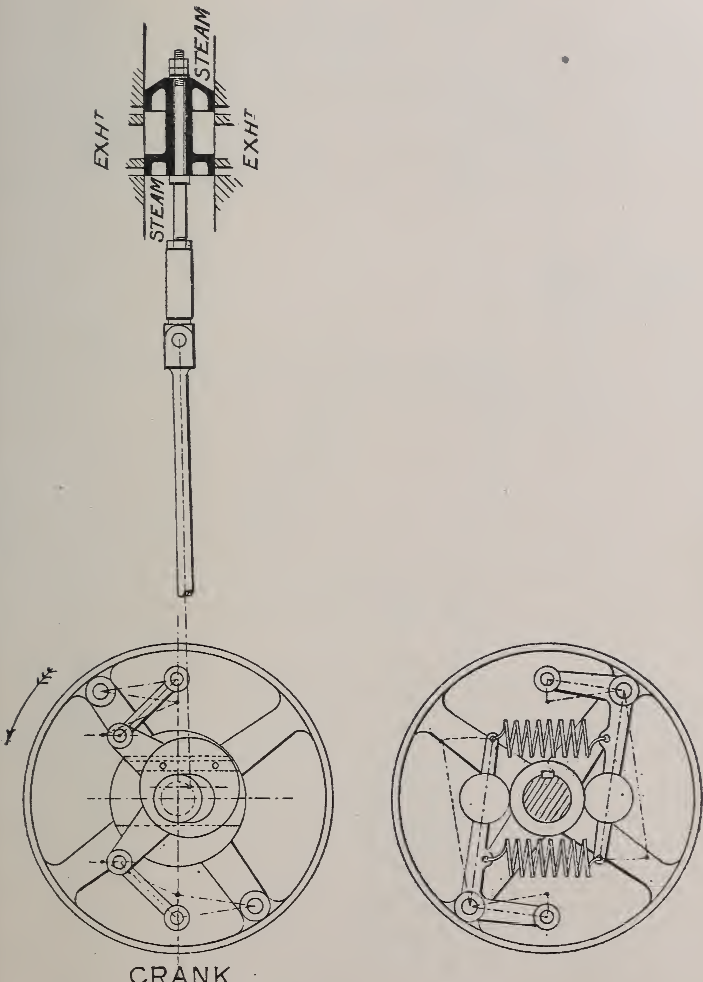


FIG. 14.

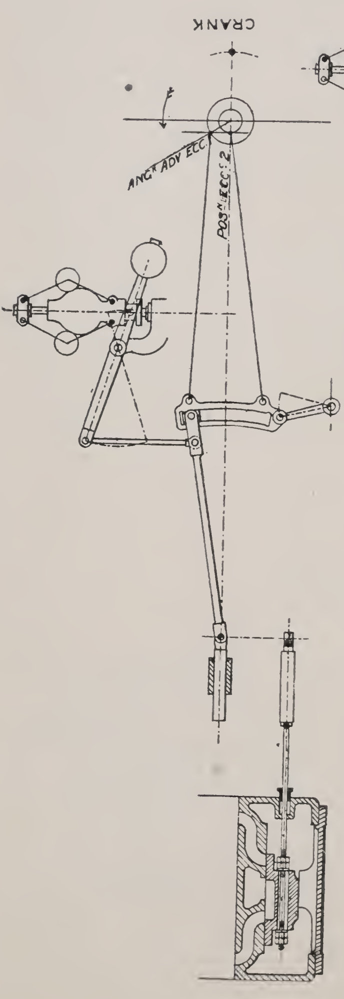

CRANK
ANG^R ADV ECC.
POS^N ECC^S 2

Fig. 15.

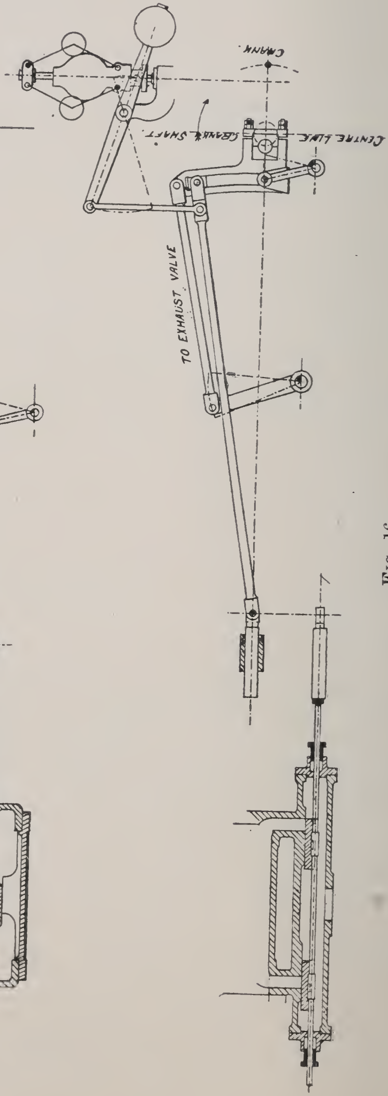

CRANK
CRANK SHAFT
CENTRE LINE
TO EXHAUST VALVE

Fig. 16.

being obtained by having two eccentrics fixed on the shaft, and giving a variable travel to the expansion valve by means of a link and die. (Example, fig. 17.)

3. By two slide valves having variable relative travel, variable angular advance of expansion eccentric, but constant negative lap of valve, the variation being obtained by having two eccentrics fixed on the shaft, and giving a variable travel and angular advance to the expansion valve by means of a compound link and die. (Example, fig. 18.)

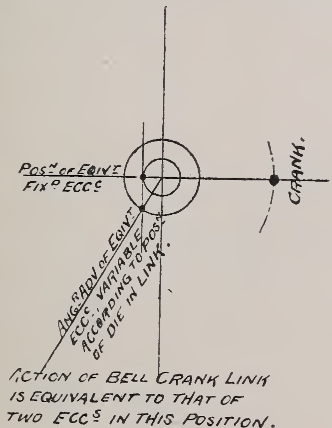


FIG. 16A.

Figure 19 is the valve diagram for this gear.

4. By two slide valves having variable lap to expansion valve, but constant relative travel and angular advance of eccentrics. The variable lap being obtained:

(*a*) By making the expansion valve in two parts, and separating them or closing them up by a right and left-hand screw on the valve spindle as in Meyer's gear, or by other means. (Example, fig. 20.)

(*b*) By making the back face of the main valve circular, with right and left-handed helical ports, and having a circular expansion valve with edges at the same angle, and increasing or diminishing the effective width of the expansion valve by turning it as in the Ryder gear. (Example, fig. 21.)

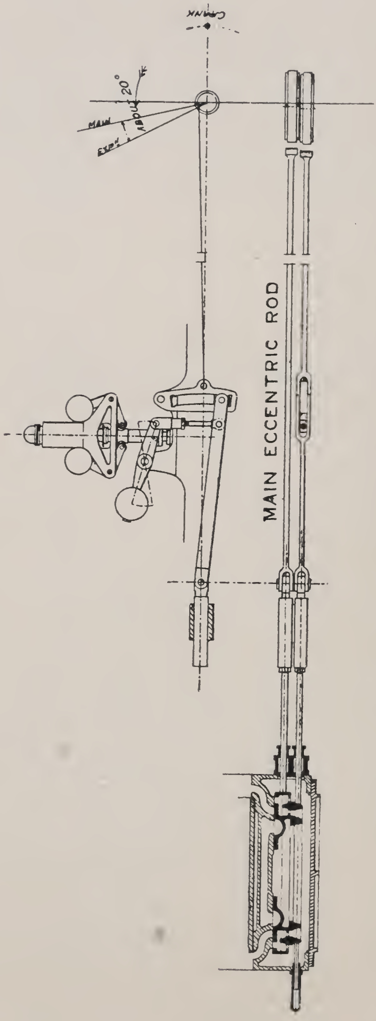


Fig. 17.

(*c*) By making the ports in the back of the main valve oblique to right and left, but the face flat, and the expansion

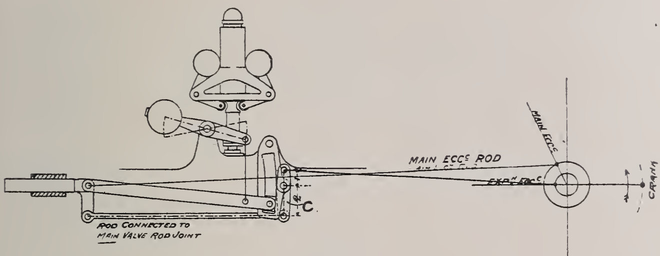


FIG. 18.

valve with edges at the same angle, and increasing or diminishing the effective width of the expansion valve by

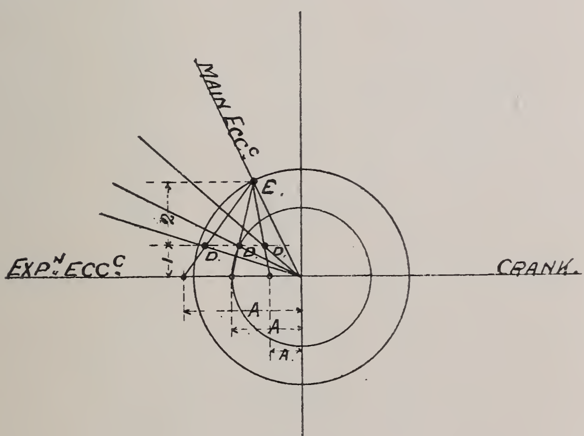


FIG. 19.

Let distances A.A.A. = half travel of expansion valve, according to position of die in link when direct coupled to eccentric rod.

Join A.A.A. to B.

Angles of advance of expansion valve will be lines drawn through A.A.A.B., cutting them at D.D.D. into two parts proportional to the lever C.

Distances D.D.D.B. give diameters of space circles or the distances that the valves move apart, and the distances of D from the centre are the actual half travels of expansion valve.

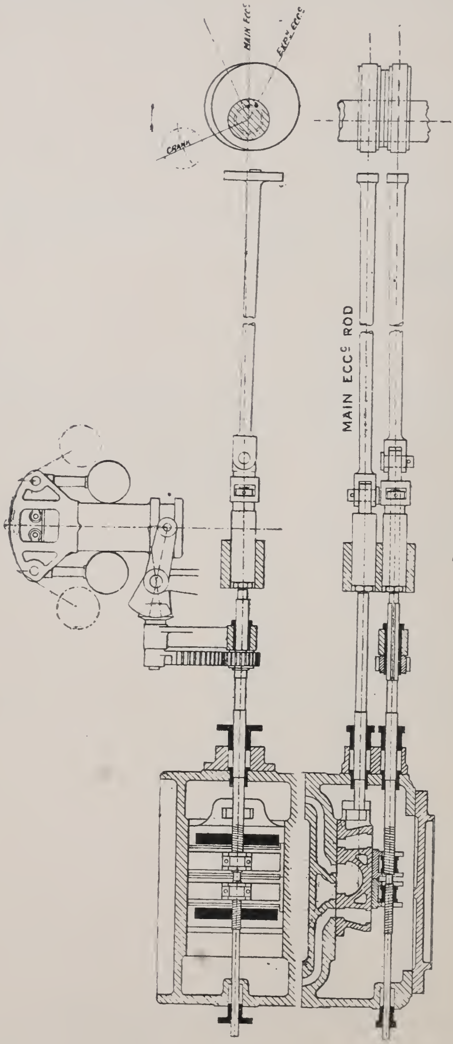


FIG. 20.

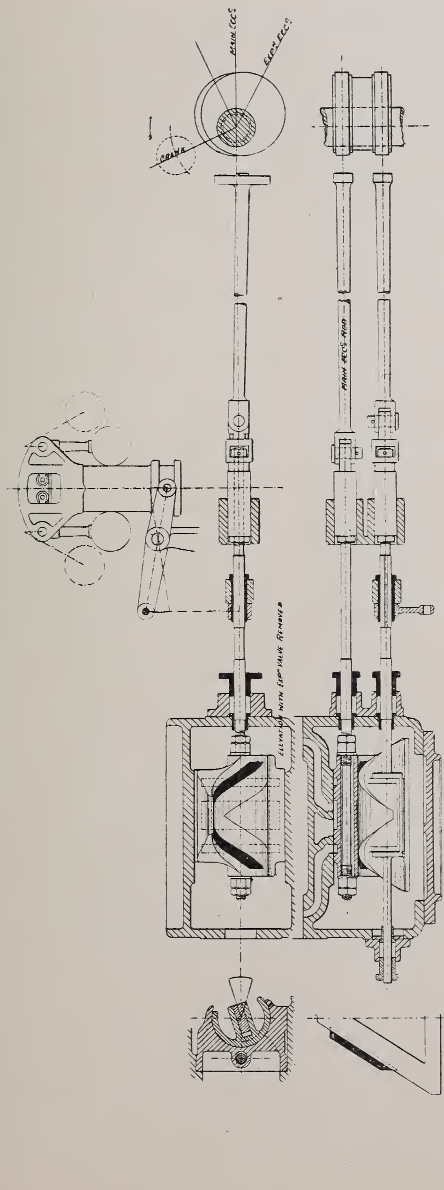

MAIN ECC.
EXPN ECC.
CRANK
MAIN ECC. ROD
ELEVATION WITH EXPN VALVE REMOVED

FIG. 21.

lowering or raising it. Fig. 22 is an example of this class of gear and is described as follows :

The valve gear consists essentially of an ordinary main slide valve A, of box section, having the usual steam ports and exhaust cavity on the front side, and two sets of inclined ports in the back, each giving admission to that end of the cylinder to which it corresponds.

The expansion slide valve B has two sets of inclined ports corresponding to the above inclined ports on back of main, and slides on a crossbar C in the steam chest, connected through a stuffing box to the governor D.

The main and expansion slide valves are separately driven direct from two eccentrics M. and E. keyed on the crank shaft; the main valve spindle is connected direct to the main valve, the expansion valve spindle being connected to the frame F, on which the expansion valve is free to slide in a direction at right angles to its motion.

Action.—The main and expansion slide valves have always the same relative travel.

The governor regulates the position of the expansion valve crosswise on the back of the main, and so moves the edges of the inclined ports nearer to or farther from the inclined ports in the back of the main valve. When the governor rises, the distance between the cutting off edges is diminished and steam is cut off earlier; when the governor falls the distance is increased, and the cut-off takes place later.

The bottom section on fig. 22, shows the expansion valve in its highest position when the distance between the cutting off edges is a minus quantity giving a positive lap, and the cut-off is at O, of the stroke. The other views show the governor and expansion valve in its lowest position when the distance between the cutting off edges is a positive quantity, giving a negative lap, and the cut-off is at $\frac{3}{4}$ of the stroke.

The following table gives a summary of the above methods of obtaining variable expansion by means of a slide or slide valves. (Table, fig. 23.)

In all the above it will be seen that the governor in order to vary the cut-off, has to change the position of the valve in relation to its driving mechanism, or, in other words, to do the work of shifting the valve on its face. The work to be

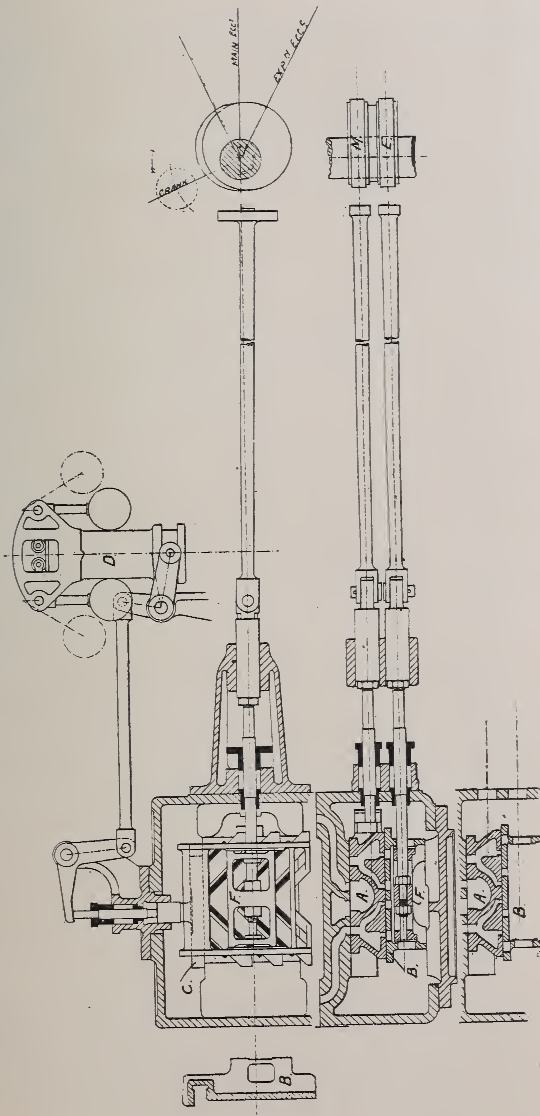


FIG. 22.

FIG. 23.—TABLE OF AUTOMATIC EXPANSION GEARS (SLIDE VALVE).

		Slide valve.	Eccentric.	Angular advance.	Travel.	Lap.	Gear.
1st	*a*	One.	One, shifting.	Varying.	Varying.	Constant.	Shaft governor. Fig. 14.
	b	One.	Two, fixed.	Varying.	Varying.	Constant.	Link motion. Fig. 15.
	c	One.	One and radial gear.	Varying.	Varying.	Constant.	Radial gear. Fig. 16.
2nd		Two.	Two, fixed.	Constant, both eccentrics.	Varying expansion.	Constant negative of expansion valve.	Link and die to expansion valve. Fig. 17.
3rd		Two.	Two, fixed.	Varying expansion eccentric.	Varying expansion	Constant negative of expansion valve.	Compound link and die to expansion valve. Fig. 18.
4th	*a*	Two.	Two, fixed.	Constant, both eccentrics.	Constant.	Varying expansion.	Meyer's gear. Fig. 20.
	b	Two.	Two, fixed.	Constant, both eccentrics.	Constant.	Varying expansion.	Rider's gear. Fig. 21.
	c	Two.	Two, fixed.	Constant, both eccentrics.	Constant.	Varying expansion.	Trapezium gear. Fig. 22.

done is usually very considerable, and necessitates a very powerful governor.

Fig. 24.—Table of Powers of Some Hartnell Type Governors with a Variation of 8 Per Cent.

Size of steam ports for 80lbs. per square inch steam pressure.	Diameter of governor balls.	Revolutions per minute. Min.	Mean.	Max.	Total variation.	Diameter of path of balls.	Total ft.-lbs. to open balls speed rising from zero to max. A.	Power of governor — ft.-lbs. speed rising from min. to max. B.
5″ × ⅝″	3¼ in.	278·4	290	301·6	8 %	6¼″ to 12½″	27	10·9
6″ × ⅝″	3½ in.	268·8	280	291·2	8 %	6¾″ to 13½″	37	15·1
7″ × ¾″	3¾ in.	259·2	270	280·8	8 %	7¼″ to 14½″	49	21·25
9½″ × 1⅛″	4⅜ in.	240·96	251	261·04	8 %	8¾″ to 17½″	97·5	40
12″ × 1¼″	4¾ in.	232·32	242	251·68	8 %	9⅝″ to 19¼″	141	56
17″ × 1¾″	5½ in.	216	225	234	8 %	11½″ to 23″	270	101

$$A = \frac{C\ \text{max.} + C\ \text{min.}}{2} \times (r\ \text{max.} - r\ \text{min.}),$$ sometimes given as power of governor.

$$B = \frac{C\ \text{max.} - C\ \text{min.}}{2} \times (r\ \text{max.} - r\ \text{min.})$$

(C = cen. force, r = rads. of path of balls in feet.)

The above table gives the power of some Hartnell type governors for different sizes of steam port and steam pressures, with percentages of variation. (Table, fig. 24.)

CHAPTER XII.

Automatic Trip Valve Gear.

In this class of valve gear the varying cut-off is obtained by causing the valve to be released or disconnected from its mechanism at varying periods of the stroke, allowing the valve instantly to close, it having been opened against a resistance as that of a spring, weight, or vacuum. This gear

is generally used in connection with Corliss valves, or in connection with valves of the Sulzer type, but is also used in connection with slide and gridiron valves.

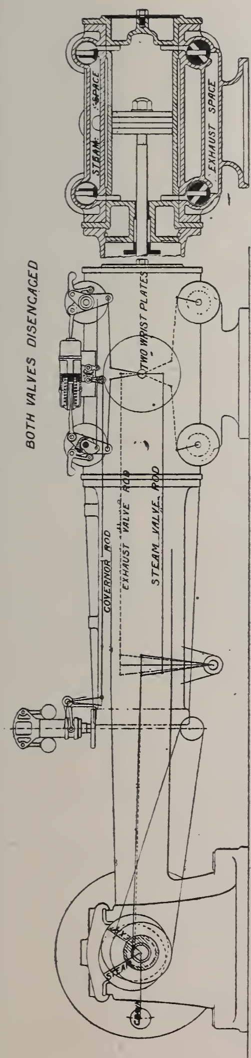
Trip gears are of two classes, viz. :—

(*a*) Gears in which the release of the valve from its mechanism can only take place when the valve is moving in the opening direction, leaving the catch loose to be put into engagement by its own weight or a spring at the commencement of the next stroke. (Examples, figs. 25 and 26.)

(*b*) Gears in which the release takes place at any time, whether the valve is moving in an opening or closing direction, and in which the catch has a positive movement, compulsorily bringing it back into engagement at the commencement of the next stroke. (Examples, figs. 27 and 28.)

The advantages of the latter class of trip gears over the former are many, chief amongst them being, first, that they permit of both steam and exhaust valves being driven from the same eccentric without limiting the range of cut-off to less than half stroke, as was the case with the early Corliss gears. Second, they allow of the engine being run at a much higher speed than do trip gears which rely upon the weight of the catch or a spring to bring it back into position for re-engagement at the commencement of every stroke.

In either case the work to be done by the governor in adjusting the mechanism to vary the time of release of the valve, and thereby the cut-off is very light, and this in itself is a recommendation for the adoption of this class of valve gear if an engine is required to run under varying loads with very small percentages of variation in speed.



Longitudinal Section of Cylinder, shcwin valves in central position.

NOTE.—Both steam valves open—neither disengaged from wrist plate.

FIG. 25.

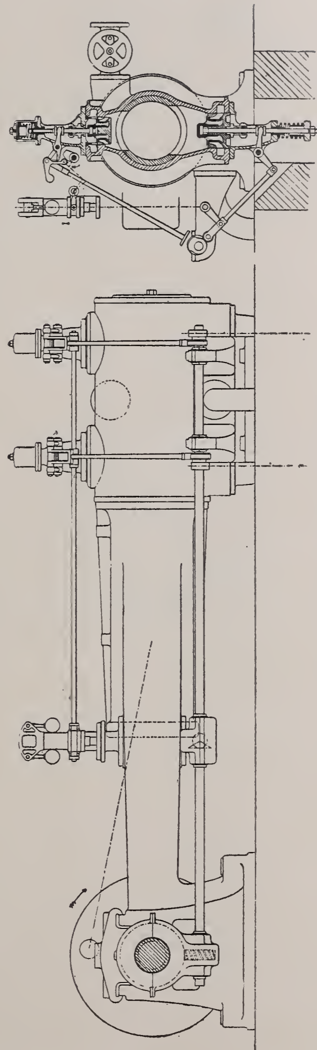


FIG. 26.—Clench and Co.'s Trip Gear.

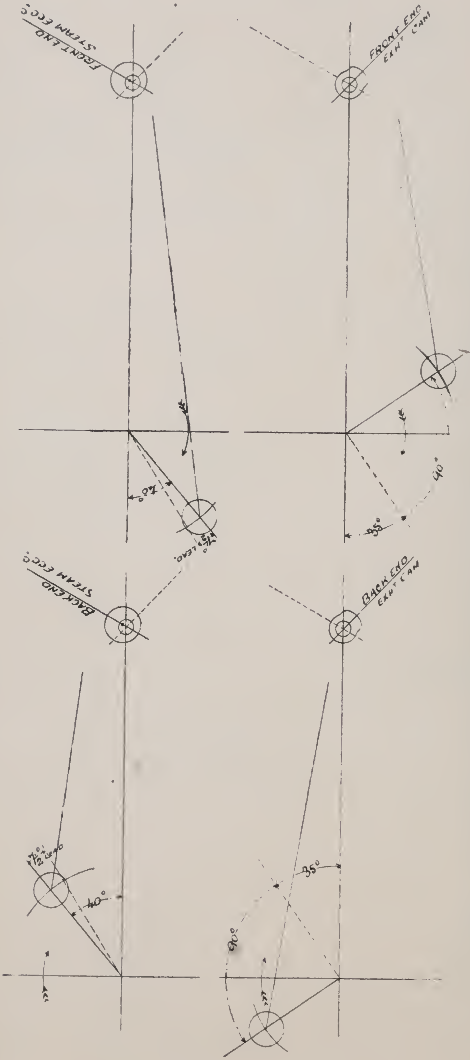


FIG. 26A.—Angles of Crank for Setting Eccentrics and Cams—Clench and Co.'s Trip Gear.

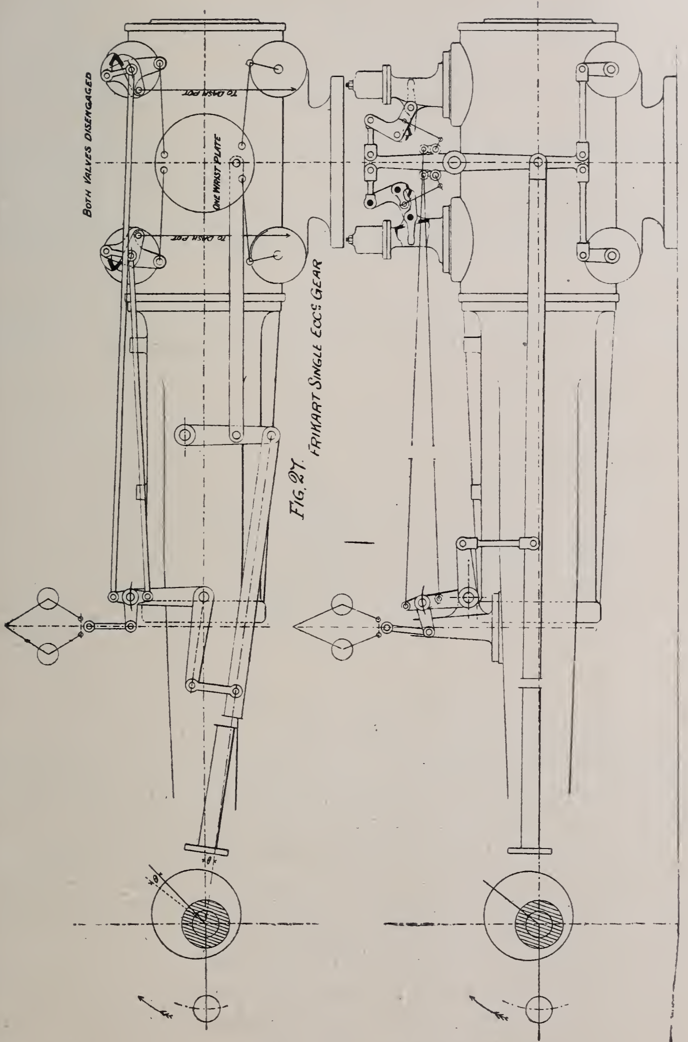


Fig. 28.— Drop Valve Single Eccentric Gear.

CHAPTER XIII.

Positions of Governor Ball and Gear for Different Points of Cut-off.

In designing a governor for any valve gear it is necessary to first determine the positions of the gear to give the required lap, travel, angular advance or whatever it may be, for a number of points of cut-off or angles of the crank, and then to find the position of governor ball for these points of cut-off.

These points should not be equal divisions of the stroke, viz., $\frac{1}{4}$, $\frac{3}{8}$, $\frac{1}{2}$, $\frac{5}{8}$, &c., such as are generally taken in laying

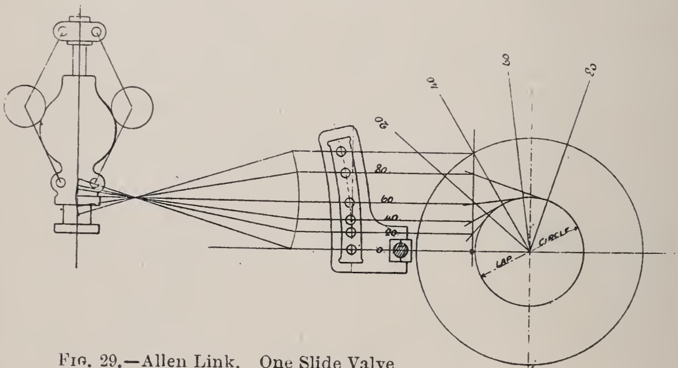


Fig. 29.—Allen Link. One Slide Valve.

down the Zeuner diagram; but should be such as will give equal divisions of the load, or, what amounts to the same thing, equal alterations of mean effective pressures.

The percentage of the stroke at which the steam must be cut off to give a mean effective pressure representing any given percentage of the maximum pressure, is given by the following table, calculated by the formula given in D. K. Clark's Mechanical Engineer's Pocket-book :—

Fraction of stroke when steam is cut off	·079	·125	·185	·25	·334	·425	·526	·64	·765
Mean effective pressures of maximum	10	20	30	40	50	60	70	80	90 per cent.

Figs. 29 and 29*a* are expanded Zeuner diagrams of link and die type valve gears, exhibiting a simple method of obtaining the proportions of link, position of die in link, and position of governor arm for different loads.

Fig. 30 shows the same for a valve gear having variable lap to expansion valve, but constant relative travel and angular advance of eccentrics. The construction of the diagram is as follows :—

On a diagram giving the position and the angular advance of the main and expansion eccentrics draw lines representing

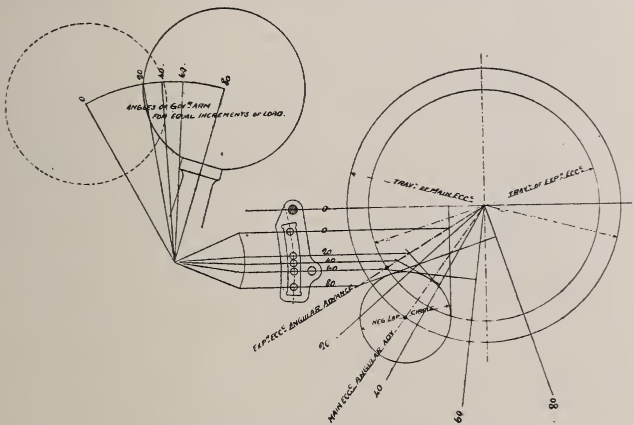


Fig. 29a.—Hartnell Gear. Two Slide Valves.

the angles of the crank for the different points of the stroke at which the steam is cut off, *and let these points be arranged so as to give equal alterations of mean effective pressures, or, in other words, equal alterations of load on the engine.*

At right angles to the lines representing the angles of the crank draw lines through the centre of the main eccentric, and on the expansion eccentric centre, touching these lines, describe semi-circles above and below the line joining the centres of the two eccentrics. The radii of these semi-circles above the line will represent negative laps, those below positive laps for the different points of cut-off.

Draw parallel lines touching these lap circles, and the distances between them will represent alterations of lap for

the different points of cut-off. If the expansoin valve port edges are at an angle of 45 deg., then these distances will also represent alterations in height or lift of the valve, and these again may represent the *lift of the governor.*

With a radius in the same proportion to the governor arm as the lift of the valve is to that of the governor, describe an arc on the lines of lift, and from the points of its intersection with them draw lines to the centre. Lines from the

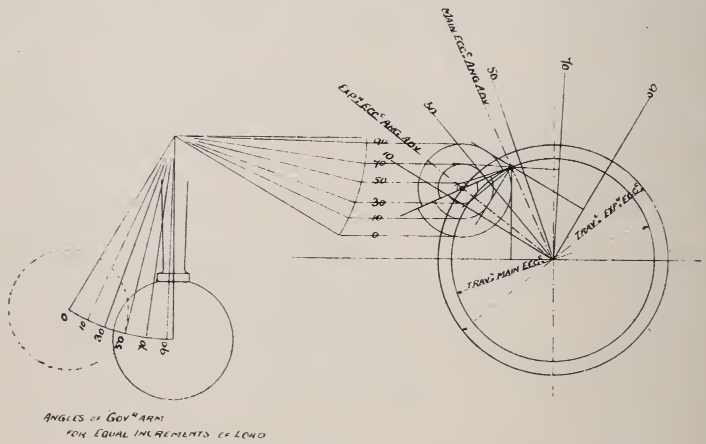


Fig. 30.—Trapezium Gear. Two Slide Valves.

same centre at an angle to these equal to that of the governor arm will give the position of the governor arm and ball for the different points of cut-off.

Positions of Governor Ball for Trip Gears.

Fig. 31 is a diagram of a trip gear of the type shown in fig. 26, page 47, giving the position of gear, shape of trip cam, and lift of governor for different loads.

The construction of the diagram is as follows :—

On a diagram giving the centres of motion of levers, direction of eccentric rod and centre of governor lifting lever spindle ; on the line of direction of motion of eccentric rod, describe a circle representing the travel of the

eccentric, and on this mark off the position and angular advance of eccentric.

From the commencement of the stroke, which is measured along the line of angular advance of eccentric,

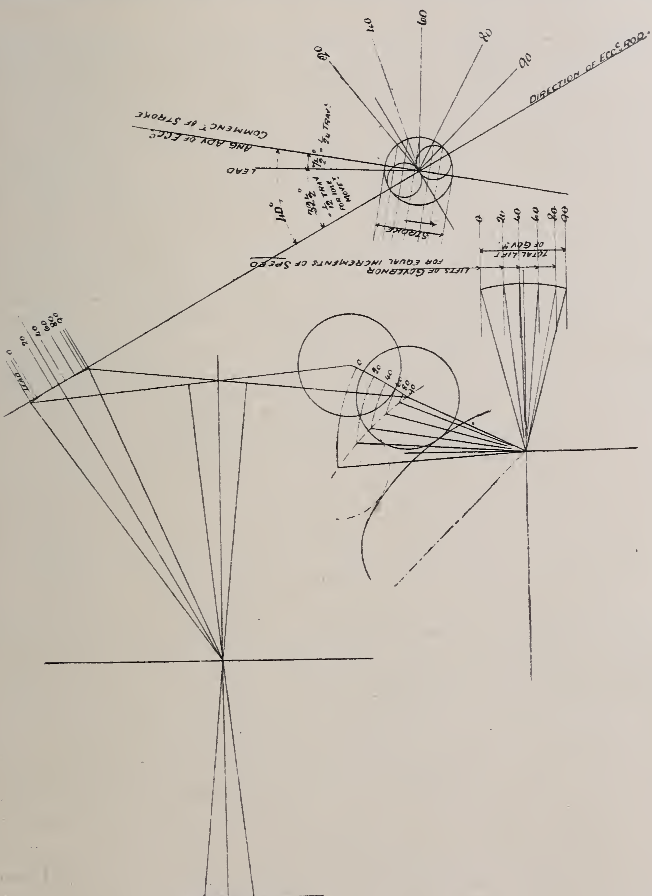


FIG. 31.

mark off the different points of the stroke at which the steam must be cut off, in order to give equal alterations of mean effective pressure, or, in other words, equal alterations

of load; and from the centre of circle draw radial lines representing the angles of the crank.

On the line of direction of motion of eccentric rod, describe the valve circles cutting the lines representing the angles of the crank.

Mark off above and below the centre line of motion of the valve lifting links, the distances cut off by these valve circles, thus giving the position of links and lever for the various points at which steam is required to be cut off.

Transfer these distances to the arc of the circle described by the roller on tail end of hanging trip piece, and from the centre of governor lifting-lever spindle describe arcs of circles passing through these points.

Next, from the same centre draw radial lines representing the lifts of the governor lever for equal alterations of speed and a curve drawn through the points of intersection of the arcs of circles with these lines, gives the shape of cam required to cause the trip to take place at any part of the stroke corresponding to position of governor.

CHAPTER XIV.

Positions of Governor Ball for Different Speeds within the Total Variation.

The next thing to do in designing a governor is to find the position of governor ball for equal alterations of speed within the total variation, and to so arrange that these shall correspond, if possible, with the different positions of governor ball required for the equal alterations of load.

Let the total variation be 2 per cent, and the number of points of cut-off or positions of the governor be taken as 5, then for the governor to be regular the variation between each of these positions should be quarter of 2 per cent, or ·5 per cent. The centrifugal force of the balls in each of these positions, and at these speeds should next be calculated, and the governor loaded accordingly.

Fig. 32 is an example of the loading of a governor of the type shown by fig. 9, page 21.

Fig. 33 is an example of a Hartnell governor, as shown in fig. 8, page 20; fig. 34 speed and load curve for the same.

Positions for the sleeve or balls for equal alterations of speed in various types of governor can be found as above,

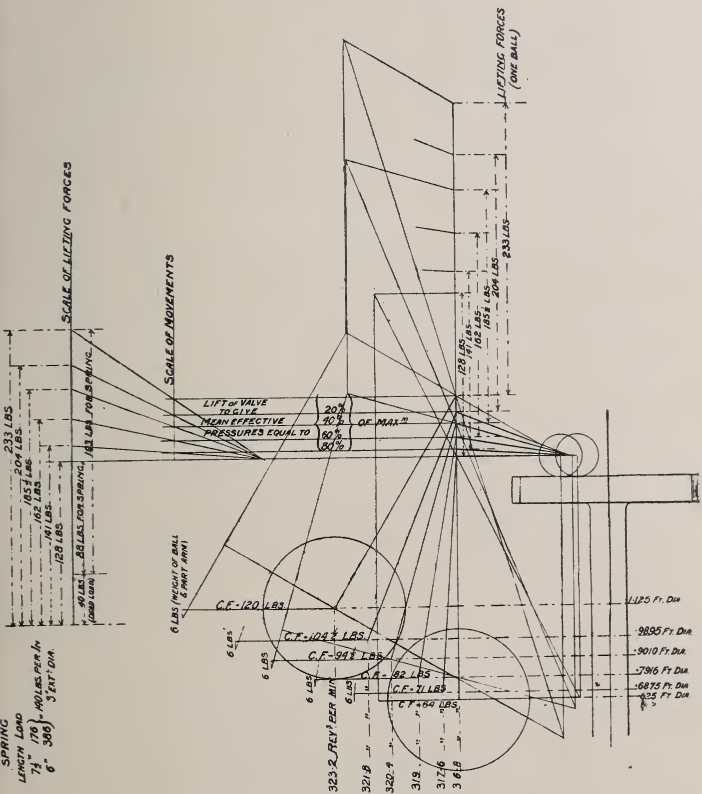


FIG. 32.

and these compared with the positions required by the various types of valve gear as obtained by the methods explained already.

It will be found impossible in practice always to so load any governor that it will run with equal alterations of speed

for the positions required for equal alterations of load by *any* valve gear, but by a careful selection a governor may

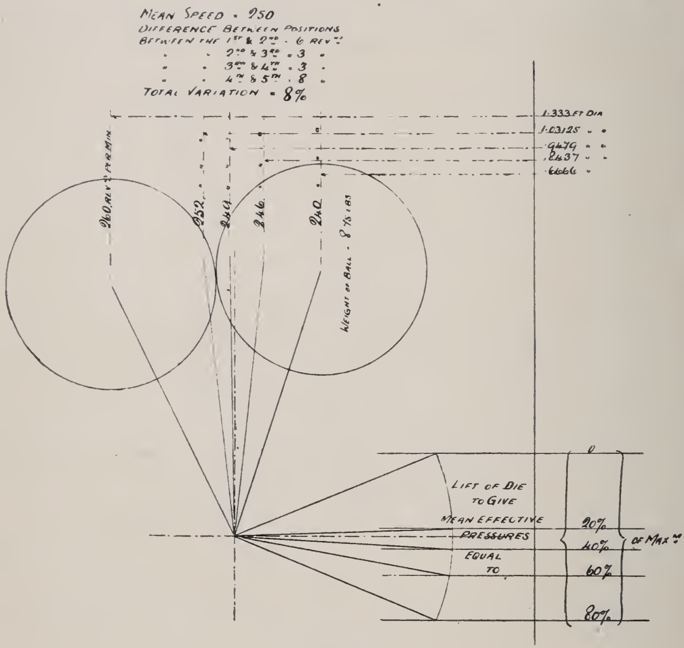


FIG. 33.

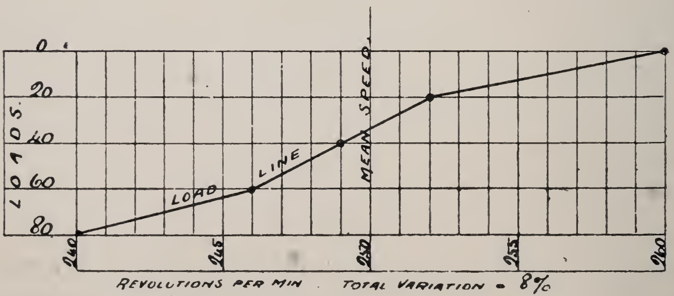


FIG. 34.

always be found, the movements of which approximately agree with the movements of the valve gear.

CHAPTER XV.

The Crank Shaft Governor.

The crank-shaft governor was invented as far back as 1868, when Messrs. Robey, of Lincoln, applied a ball governor, placed horizontally, on the crank shaft of an engine to Dodd's wedge motion for shifting the eccentric, thereby obtaining auto-expansion with one slide valve having variable travel and angular advance of eccentric, but constant lap.

Fig. 35 shows Robey's adjustable eccentric, worked by a ball governor and Dodd's wedge motion.

In designing a shaft governor there are four main points that should be taken into consideration over and above what has been said in the previous chapters in regard to governors generally.

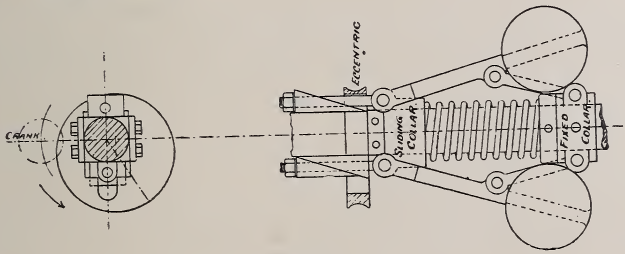


Fig. 35.

1. On account of the power of a governor depending more upon the speed than upon the weight of the balls, as was explained previously, the crank-shaft governor is not suitable for engines running at less than, say, 200 revolutions per minute.

2. From its peculiar arrangement on the crank shaft of an engine, which lays horizontally, the governor must be a pure spring loaded governor, and should be arranged in such a way that its parts are unaffected by gravity.

3. The parts should also be arranged in such a way that they are as nearly as possible unaffected by inertia forces,

due to change of speed, unless it is intended to utilise such forces to operate the governor as in the case of the inertia governor to be described in the next chapter.

4. The attainment of perfect balance of the revolving parts in a shaft governor is not so simple a matter as it is in the case of an ordinary upright governor, and yet on account of the governor being used on high-speed engines only it is a matter of even greater importance.

The shaft governor is used both as a throttle valve governor as shown in fig. 10, page 22, and as an automatic governor, as shown in fig. 14, page 35.

CHAPTER XVI.

The Inertia Governor.

It was said in a previous chapter that centrifugal force was that on which almost all governors depend for their action; but there is another force which may be, and has been, taken advantage of to bring about a change in the position of the balls in a governor.

This force is that due to the inertia of a revolving body apart from its centrifugal force.

In all the governors considered in the previous chapters the balls are connected to the driving spindle in such a manner that they must turn with it, and any increase or decrease in speed of the spindle, and, consequently, of the engine, is transmitted immediately to the balls.

They are only free to move in a direction outwards from the spindle and to describe a larger or smaller radius; always, however, turning with the same angular velocity as the spindle.

Conceive, now, that the balls of a governor are connected to the spindle in such a way that at the same time as they have a movement outwards from the spindle they are constrained to turn around it to a small extent, *in a direction contrary to that in which they revolve.*

Then the driving force of the spindle, in tending to drive the balls, will be resisted by their inertia, and any increase in speed of the spindle will not be transmitted immediately to the balls.

Some of the force which would have been expended in increasing the speed of the balls will, instead, have been spent in tending to open them out; while the increase of centrifugal force which would have acted to do this, had the increased speed of the spindle been immediately transmitted to the balls, is lost and does not come into play until the balls have acquired the increased speed of the spindle.

It may be here mentioned that the force which has acted to open out the balls, and which is measured by the resistance it meets with in their inertia, acted only during the change of speed; also that the increase of centrifugal force which otherwise would have acted, would continue to act, and does act after the change of speed has been effected.

Let the weight of the balls in an inertia governor be 10 lb. and the radius of the path described by them be from 3 in. to 6 in.; also let the speed increase from 300 to 303 revolutions per minute, or 5 to 5·05 revolutions per second.

The energy, viz., $\frac{W v^2}{2 g}$ stored up in the balls when revolving at the larger radius and at a speed of 303 revolutions per minute or 5·05 revolutions per second is 39·25 ft. lb.; and that stored up in the balls when revolving at the smaller radius, at a speed of 300 revolutions per minute or 5 per second, is 9·6 ft. lb.

Then the work done by the force which has acted during the change of speed against the inertia of the balls in tending to open them out is represented by $39·25 - 9·6 = 29·65$ ft. lb.

The centrifugal force, viz., $\frac{W v^2}{gr}$, exerted by the balls when revolving at the larger radius at a speed of 303 revolutions per minute or 5·05 per second is 157 lb. And the centrifugal force exerted by the balls when revolving at the smaller radius at a speed of 300 revolutions per minute or 5 per second is 77 lb.

The increase of centrifugal force would have been 80 lb., and the mean of this multiplied by the distance through which the balls have been moved, viz., $\frac{80}{2} \times \cdot 25 = 10$ ft. lb.

Thus the work done by the force which has acted during the change of speed against the inertia of the balls in tending to open them out is much greater than that which would have been done by centrifugal force.

But the *force* which has acted to produce this greater quantity of work depends on the time during which it has taken to act, or, in other words, during which the change of speed has taken place.

It is possible for the change of speed to take place so slowly that this force becomes inappreciable, and therefore of no value in opening out the balls of a governor.

Let the change of speed in the governor at present under consideration take place in one second, that is, while the governor is making approximately three revolutions.

What is the force that, acting for one second on a weight of 10 lb., will produce an acceleration corresponding to this change of speed, viz., $15 \cdot 85 - 7 \cdot 85 = 8$ ft. per second?

If the acceleration had been $32 \cdot 2$ ft. per second, then the acceleration would be that which would have been produced by gravity, viz., the weight's own weight in falling for one second.

But this acceleration is only 8 ft. per second, or one quarter that produced by gravity; therefore, the force required to produce it would be only one quarter the weight, viz., $2\frac{1}{2}$ lb.

And the space described by a weight, already moving at the rate of $7 \cdot 85$ ft. per second, in acquiring an added velocity of 8 ft. per second, is $\frac{15 \cdot 85 + 7 \cdot 85}{2} = 11 \cdot 85$ ft.

This $11 \cdot 85$ ft. $\times$ $2\frac{1}{2}$ lb. $= 29 \cdot 62$ ft. lb., as before.

Now, let the change of speed take place in two seconds, then the mean force acting would be $1\frac{1}{4}$ lb.; in three seconds $\cdot 625$ lb., and so on.

For the mean force to amount to as much as the mean centrifugal force, which would have acted, viz., 40 lb., the change of speed would have to take place in $\frac{1}{16}$th of a second.

In constructing the inertia governor it is usual to considerably reinforce the inertia of the balls by the addition of other revolving weight.

Let the inertia of the balls in the governor under consideration be reinforced by the addition of other revolving

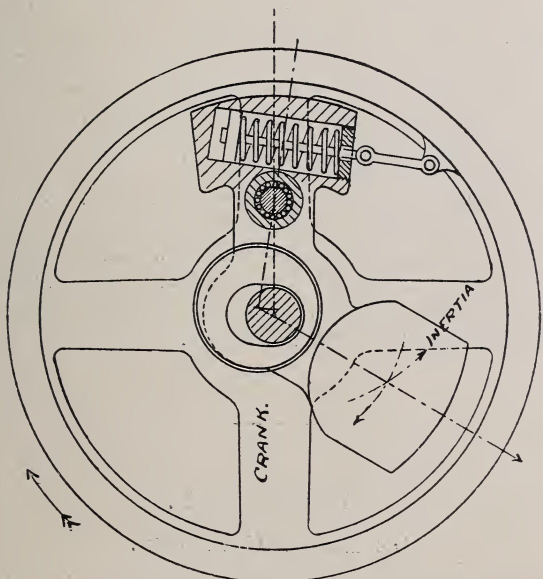


FIG. 36.

weights to the extent of, say, 16 times = 160 lb. Then the force acting to produce an acceleration of 8 ft. per second *in one second* would be equal to 16 × 2·5, viz., 40 lb.

Thus, if it is required that the force acting in an inertia governor, when the change of speed takes place is one second, shall be equal to the centrifugal force due to the weight of the balls, the total weight in the governor,

revolving at the same radius as the balls, must equal 16 times that of the balls alone.

The idea of the inertia governor was first embodied by Siemens, previous to the year 1866, in his differential governor.

Siemen's differential governor is described by Goodeve, who says:—

"The chief peculiarity of the differential governor consists in the fact that the *whole energy* stored up in the revolving balls is ready to act upon the steam valve at the first instant that the engine attempts to vary from the pendulum, whereas in the ordinary governor the *additional energy* stored up in the balls by increased velocity of rotation is the power available to control the valve.

"One action is slow and comparatively feeble, the other is instantaneous and cannot be resisted."

Fig. 36 is an example of an inertia governor. Up to the present the application of this principle seems to have been confined to the crank-shaft governor, but there seems to be no reason why it should not be applied to any ordinary upright governor, the load or the counterpoise being made to serve the purpose of inertia weight.

CHAPTER XVII.

Gas Engine Governors.

While the steam engine governor depends for its action on a general change of speed of the engine, and should not be affected by the momentary change of speed, or what has been called the jerkiness of the engine, the gas engine governor, on the contrary (as usually constructed), depends entirely for its action on this momentary change of speed or jerkiness of the engine. In fact, the greater the momentary change of speed, the greater the action of this governor in cutting out more impulses to reduce the power of the engine to suit its lighter load.

Thus: In a gas engine governed in the usual way, *i.e.*, on the hit and miss principle, the less the load the more unsteady will be the running of the engine.

Figure 37 shows a centrifugal governor as applied to gas engines made by Messrs. Crossley. Its action is as follows: At every firing stroke of the engine the governor G rises and lifts the flat plate P suspended from the lifting levers, from between the plate V on the gas valve lever and the head of the gas valve spindle S.

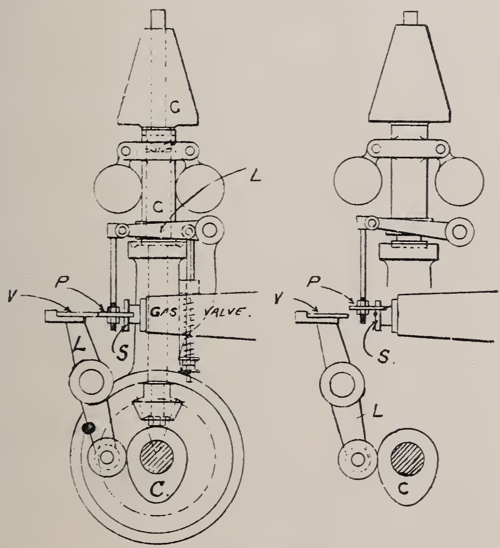


FIG. 37.

At full load the fly wheel prevents the impulse given by the firing stroke from causing the momentary speed, and consequently the governor, to rise too high; therefore, by the time the cam C is ready to operate the lever L, the plate P is again in position between the plate on the gas valve lever and the head of the gas valve spindle. Should the load on the engine, however, be diminished, the impulse of the firing stroke causes the momentary speed and, consequently, the governor to rise so high that by the time the cam is

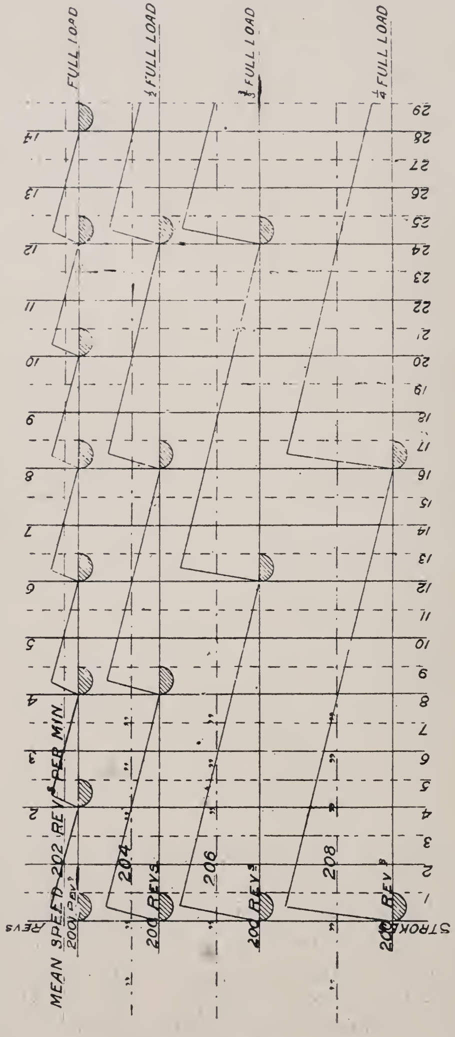


FIG. 38.

Diagram showing periodical fluctuation of speed at various loads for an Otto gas engine with flywheel sufficiently heavy to prevent more than 2 per cent periodical fluctuation at full load.

ready to operate the lever L the plate P is not again in position, and the lever fails to open the gas valve S until such time as the speed has fallen to that at which the governor runs in its normal position.

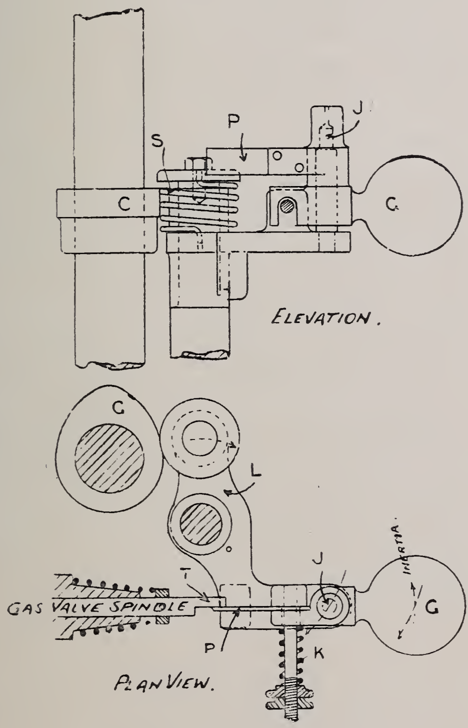


FIG. 39.

The governor is a dead-weight governor, but the sleeve is not in contact with the counterpoise until the engine has reached a certain speed, thus enabling the sleeve to rise quickly as the engine gets up speed. When the normal speed is reached, the governor begins to operate as just described.

Fig. 38 is a diagram showing the cycle of operations for an Otto Cycle Gas Engine at various loads, also showing the necessary fluctuations in the speed of governor and engine.

This principle of action of the gas engine governor depending, as it does, on a continual series of sudden bounds (the less the load the greater the bound), enables another form of force, viz., inertia, to be satisfactorily taken advantage of in the operation of a gas engine governor.

Fig. 39 shows an inertia governor, as applied to gas engines, made by Messrs. Crossley. Its action is as follows: The cam C operates the lever L, the roller on which is held hard against the cam by the spring S. On the lever L, which also operates the air valve, is pivoted at J the governor weight G, and this is prevented, when the engine is running at full load, from being left behind the lever L by the spring K. To lever L is attached the striker plate P.

The spring K is so adjusted that when the load on the engine is diminished the comparatively greater impulse of the firing stroke, acting against the inertia of the weight G, causes it to be left behind, the striker P then missing the gas valve T until the speed again falls to the normal.

A more satisfactory form of governor for gas engines, considered with a view to obtaining regularity of speed merely, would be one that operated on the same lines as the steam engine governor, viz., admitting a larger or smaller amount of a previously mixed charge to the combustion chamber at every charging stroke.

CHAPTER XVIII.

Relay Governors.

The previous chapters have dealt with governors acting on the simple principle of the Watt governor, which is, that when the greater demand made on the engine for power (which occasioned the fall of speed and, consequently, the greater opening of throttle valve) is *maintained*, the fall of speed must also be maintained, and *vice versâ.*

But if, when the greater demand is made on the engine for power, occasioning a fall of speed, and consequently a greater opening of throttle valve, some other agency, apart from that of the engine itself, be introduced to regulate the length of connections between governor and valve, then the greater opening of throttle valve occasioned by the fall of speed may be maintained *without the governor having to remain in its lower position or the engine having to continue running at its lower speed*, and *vice versâ*.

This is the principle of the relay, auxiliary or supplementary governor.

Fig. 40 shows an example of a relay governor, provisionally protected by the writer, in which the agency employed to regulate the length of connections between governor and valve consists of the momentum and inertia stored up in a revolving wheel. It will be noticed that the inertia of the revolving wheel, in this case, is not used as it is in the ordinary inertia governor, to open out the balls, but is introduced to regulate the length of connections between governor and valve, the governor itself remaining to all intents and purposes a simple ordinary centrifugal governor. With this difference, however, that, instead of rising or falling as the speed of the engine rises or falls, it simply *tends* to rise and fall—this *tendency* forming the resistance against which the momentum or inertia of the flywheel acts in order to effect the alteration in the length of valve rod to give a greater or less admission of steam to suit the altered demand made on the engine for power.

It will be seen that—

1. No change can be made in the load on the engine without a change of speed of the governor tending to make it rise or fall, and making it run faster or slower than the loose flywheel; and

2. No change can take place between the speed of the governor and the speed of the loose flywheel without an adjustment being made in the length of the valve rod, immediately bringing back the engine to its proper speed, viz., that at which the governor has no tendency to rise or fall.

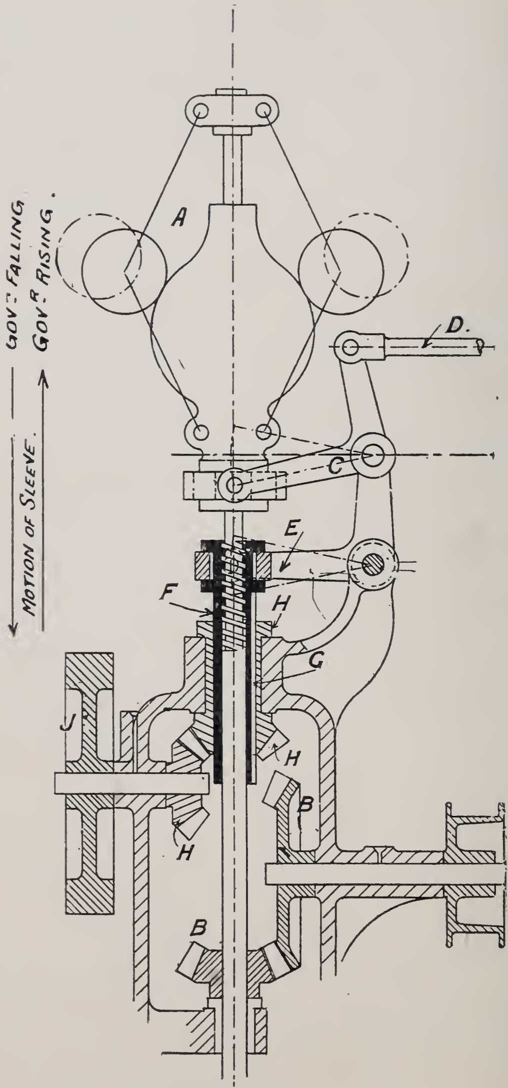


FIG 40.

A is the governor pure and simple, carried on a spindle driven by bevel wheels B B, from the engine in the ordinary way. C is a forked bell-crank lever, taking the place of the usual lifting levers, and connected direct to the throttle valve or valve gear by the rod D. E is a second bell-crank lever, which carries on one end the aforesaid bell-crank lever C, and by its movement effects a virtual alteration in the length of the rod D. F is an internally screwed sleeve, screwing on to the governor spindle and revolving with it, being prevented from screwing up or down by the fork end of lever E.

G is a feather on the outside of sleeve F, causing it to turn with the bevel wheel H through the boss of which it

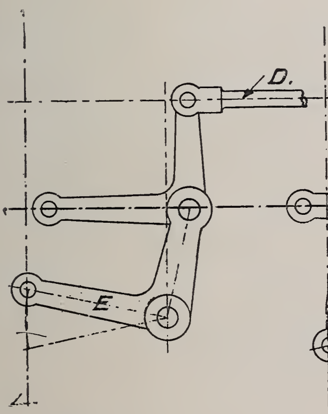


Fig. 41.

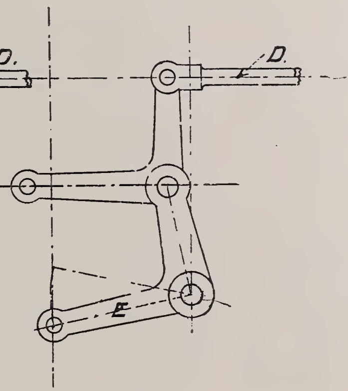


Fig. 42.

slides. J is a small flywheel, carried in a bracket bolted to the governor stand, and is geared to the sleeve F by bevel wheels H H.

As the governor rises (speed accelerating) it screws up the end of lever E, and brings the bell crank into position (shown in fig. 41), altering the position of rod D, and closing the throttle valve.

As the speed falls the governor screws down the end of the lever E, and brings the bell-crank levers into position

(shown on fig. 42), altering the position of rod D, and opening the throttle valve.

It will be noticed that the governor, after it has reached its proper speed, remains at the same height, and consequently runs at the same speed in both positions.

Fig. 43 is another arrangement of the same governor.

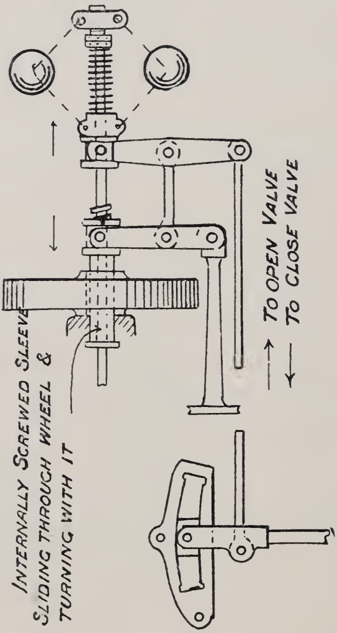


FIG. 43.

Fig. 44 shows the Knowle's supplementary governor, constructed by Messrs. Hick, Hargreaves and Co., in which the agency employed to adjust the length of connections between the governor and valve consists of the centrifugal force of another governor.

The sleeve of this supplementary governor is provided with an upper and lower flange, between which is placed a small friction wheel, running on a spindle. On the other

end of this spindle is a grooved pulley which drives, by means of a cord, a similar pulley on the adjusting nut connecting the two ends of the rod between the main governor and valve.

On the governor tending to rise or fall, it turns the friction pulley in one direction or the other, and so operates the adjusting nut, shortening or lenthening the connection between main governor and valve.

In calculating the loading, &c., of this class of governor, a variation of speed must be allowed for just as in the case

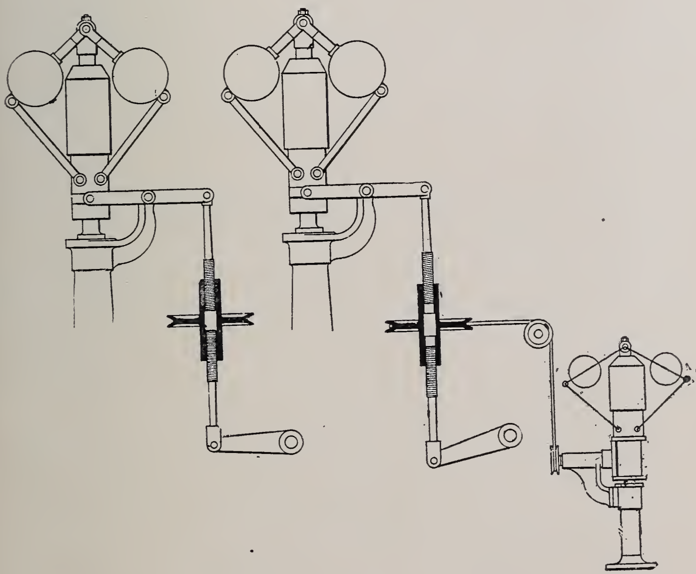


Fig. 44.

of an ordinary governor, as it is from this that the governor derives its power.

Instead of the variation being permanent, however, as in the case of an ordinary governor, it is in this case merely momentary.

APPENDIX I.

Governor Power.

"*That governor, therefore, had the greatest lifting and forcing power which, when revolving at a speed corresponding to its lowest position, would require the greatest force to make it take up the highest position at the same speed, and, conversely, when revolving at a speed corresponding to its highest position, would require the greatest force to make it take up the lowest position.*"—J. Richardson, in reply to discussion on his paper on "The Mechanical and Electrical Regulation of Steam Engines." Proc. Inst. Civil Engineers, vol. cxx., May, 1895.

The writer has found that widely different notions are held by different engineers as to what may be taken to be the power of a governor.

In the preceding pages the power of a governor is taken to be the mean *difference* of the centrifugal force at the governor's top and bottom speeds multiplied by the range, and not the mean total centrifugal force multiplied by the range which is taken by some to be the power of a governor, but which is the "total stored energy."

Of course, what is meant by the power of a governor is its "power of overcoming external resistances," and a little consideration will show that while the "total stored energy" in a governor may be very great, yet very little or none of this may be available for the power or purpose mentioned, viz., that of overcoming external resistances.

Let anyone try to lift the governor of a steam engine by a lever fixed to the lifting spindle *before the engine has reached its proper speed*, and he will find it hard; but let him try after the engine has reached its proper speed, and he will find it easy.

Let him hold the lever down after the engine has reached its proper speed, allowing it to run still faster, then loose

the lever gradually, the balls opening out against the pull of his hand, until they have taken up the position due to their top speed; the external work done by the governor will be the pull he was exerting multiplied by the distance through which that pull acted, and this is the power of the governor to overcome external resistances. (It is assumed for the moment that the governor has no valve gear, etc., to lift.)

Now let the steam be shut off and the governor still supported in its top position after the engine has stopped, then gradually lowered; the pressure exerted in doing this, multiplied by the distance through which it was lowered, will be equal to the total stored energy, or the total work which was done by the governor in lifting through its full range of movement.

The work done in compressing a spring is equal to the mean force exerted multiplied by the range through which it acts; and this is the total work done by a spring-loaded governor in lifting through its full range of movement.

Also the work done in lifting a dead weight is the weight multiplied by the height through which the weight is lifted; and this is the total work done by a dead-weight loaded governor in lifting through its full range of movement.

But this is not the power that can be given out by a governor when running between its top and bottom speeds.

In a steam engine the mean of the pressures on the two sides of the piston, multiplied by the distance through which the piston is moved, would not be the power of the engine; neither would the mean of the tensions on the tight and slack side of a belt over a pulley, multiplied by the movement of the belt, be the power transmitted by a belt.

The power transmitted in either case would be the mean of the *difference* of pressures or tensions multiplied by the distance moved through. It may be taken that while the forces in a governor are balanced, it can have no power of overcoming external resistances, in the same way as it is taken that while the forces or steam pressures on either side of the piston of a steam engine are balanced, *it* can have no "power."

What is required in order that a governor may have "power" (supposing the forces to be already existing, viz., the centrifugal force balancing the resistance of the spring or weight) is that the balance shall be upset; in the same way as for an engine to have power, it is required that the pressure on the one side of the piston must be greater than that on the other.

The way in which the balance of pressures or forces in a governor is upset is by an increase or decrease of speed. And the amount by which it may be upset depends on the amount by which the speed may be increased or diminished; or, in other words, depends on the variation of the governor.

The power of a governor, then, may be defined as *the unbalanced force capable of being generated in a governor by a given change of speed, multiplied by the distance through which this unbalanced force can act.*

With a given variation, what is the amount by which the balance of forces in a governor can be upset?

Let a governor be running with its balls in their top position at its top speed, say, 303 revolutions per minute, and let the spring in the governor, in balancing the centrifugal force due to this speed, be exerting a force of, say, 100 lb., then while the governor continues to run at this speed it is exerting no power in the sense of overcoming external resistances.

But let the speed be suddenly reduced to the minimum, say, 300 revolutions per minute, at which it would be exerting a centrifugal force of, say, 80 lb., then the governor can exert force and do work in overcoming external resistances until the balls fall and the pressure exerted by the spring is 80 lb., balancing the centrifugal force of the governor running at its lower speed with the balls in their lower position.

Again, let a governor be running with the balls in their lower position, at a speed of 300 revolutions per minute, the spring in the governor exerting a force of 80 lb., equal to the centrifugal force due to this speed; then, while the governor continues to run at this speed, it is exerting no power in the sense of overcoming external resistances.

But let the speed be suddenly increased until the governor is exerting a centrifugal force of 100 lb., then it can exert

force and do work in overcoming external resistances until the balls open out and the pressure exerted by the spring is 100 lb., balancing the centrifugal force of the governor running at its top speed with the balls in their top position.

Let the range of this governor be 2 inches, then the work that may have been done by the governor against external resistances is $\frac{100 - 80}{2} \times 2$ in. $= 20$ in. lbs.

But the total stored energy when the governor was running in its top position was $\frac{100 + 80}{2} \times 2$ in. $= 180$ in. lbs.

Now, let the variation be increased so that the centrifugal force varies from 60 lb. to 120 lb., then the external work the governor may be capable of doing is $\frac{120 - 60}{2} \times 2$ in. $= 60$. in. lbs.

But the total stored energy, viz.:

$\frac{120 + 60}{2} \times 2$ in. $= 180$ in. lbs., is still the same.

APPENDIX II.

Examples of Governors and Governing Mechanism by various Makers.

Note.—The following illustrations and particulars have been kindly supplied by the editor of *The Practical Engineer* :—

I. *Hartnell's Patent Automatic Expansion Gear.*—Fig. 45 shows this governor and gear as made by Messrs. Marshall, Sons and Co. Limited, Gainsborough, and is described by them as follows :—

"This arrangement consists of a highly sensitive and powerful governor, of special construction, acting through a link and die on to an expansion cut-off valve working at the back of the main slide valve; the ordinary throttle valve is dispensed with, and the speed of the engine thoroughly controlled by means of the expansion valve, which regulates the admission of steam into the cylinder exactly in proportion to the duty performed by the engine.

"This gear is very simple and most reliable in its action, automatically regulating the speed with every varying load, and is thoroughly adapted for either heavy or light work. Engines equipped with this governor develop more power with greater economy in fuel than if fitted with throttle valves in the usual way."

Messrs. Marshall have now over 3,900 engines at work with this gear, with very satisfactory results.

II. *Galloway's Patent Parabolic Governor and Automatic Gear.*—This gear, illustrated by fig. 46, consists of the makers' well-known parabolic governor acting through an Allen link to give a varying cut-off, by varying the travel and angular advance of a single slide valve. Fig. 47 is a clearer view of the governor itself. The engine is fitted with separate exhaust valves operated by a fixed eccentric, giving constant compression and release, which cannot be

obtained when this mode of governing—viz., by varying travel and angular advance—is applied to a single slide valve, which also controls the exhaust.

This system has all the advantages of the two-slide valve gear.

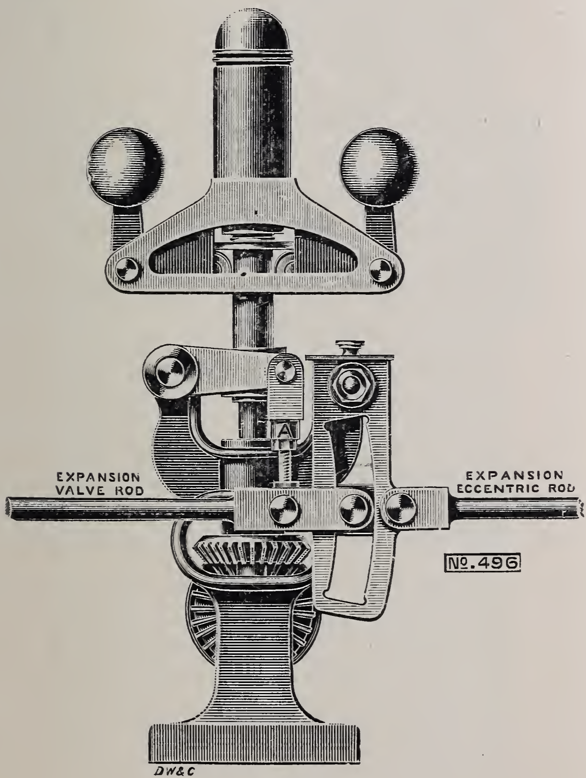


FIG. 45.

III. *Tangye-Johnson Automatic Gear.*—This gear, illustrated by fig. 48, is an "automatic" slide-valve gear, but differs from any of those described in the preceding pages in that the varying cut-off is not obtained by any ordinary action of the slide valve, such as variation of lap, travel, or

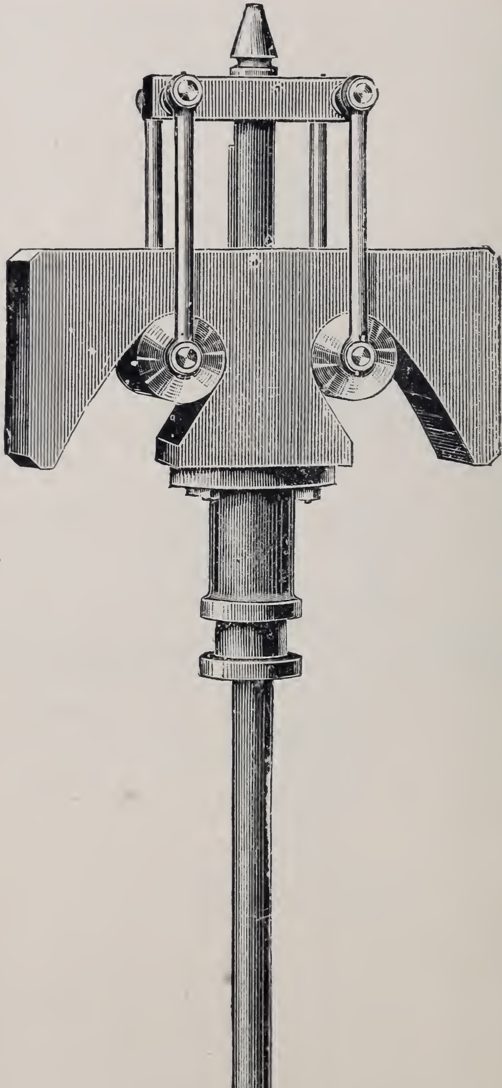


Fig. 47.

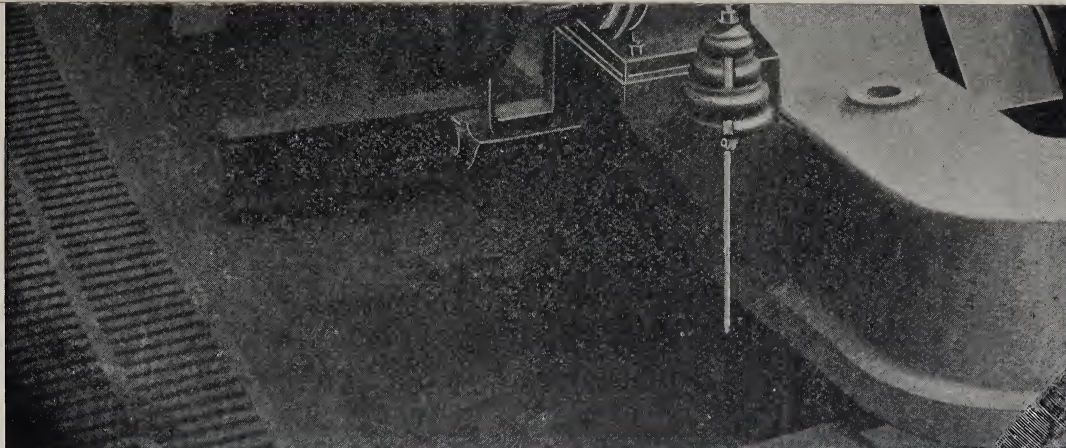


Fig. 46.

GM

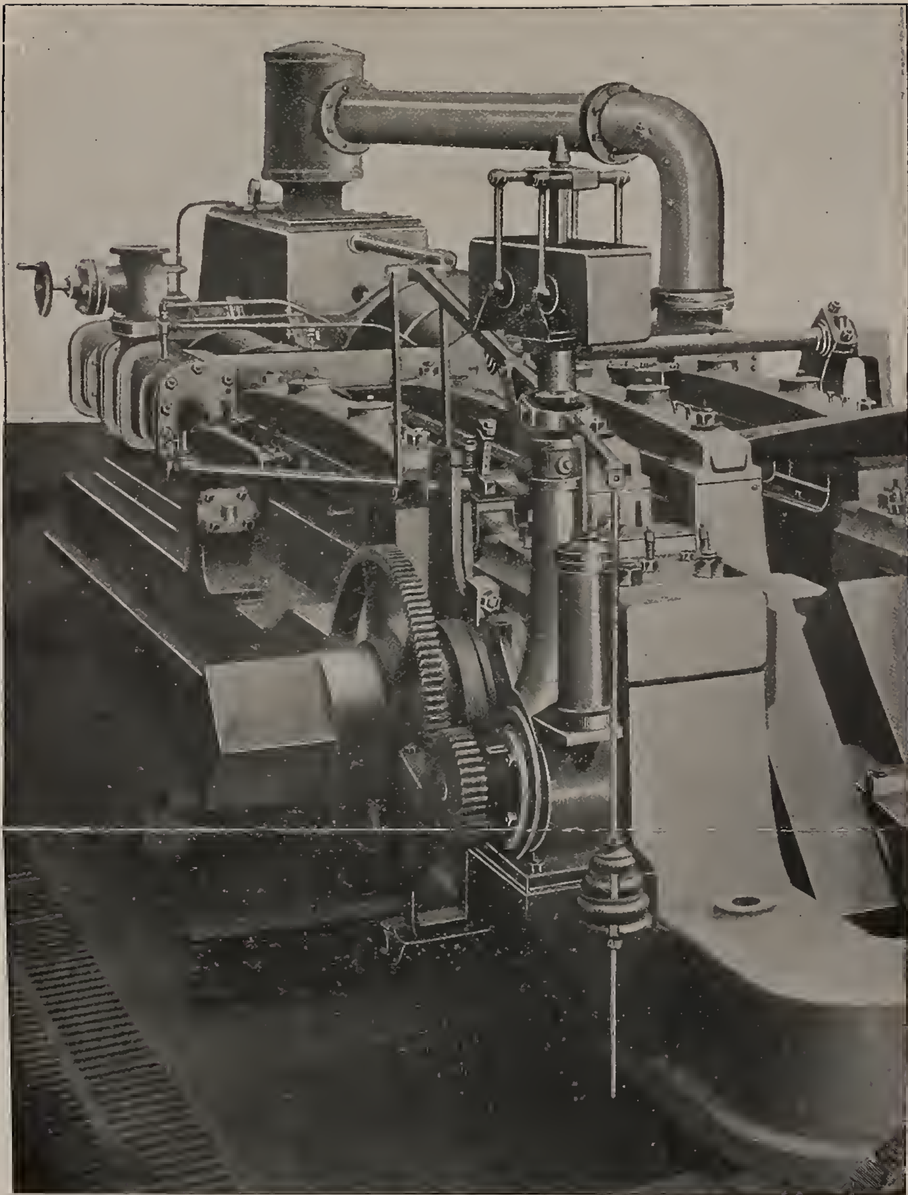


Fig. 46.

GM

angular advance, but by the rocking action of the back part of the valve, which is hinged so as to close first the port in the back of the valve communicating with one end of the

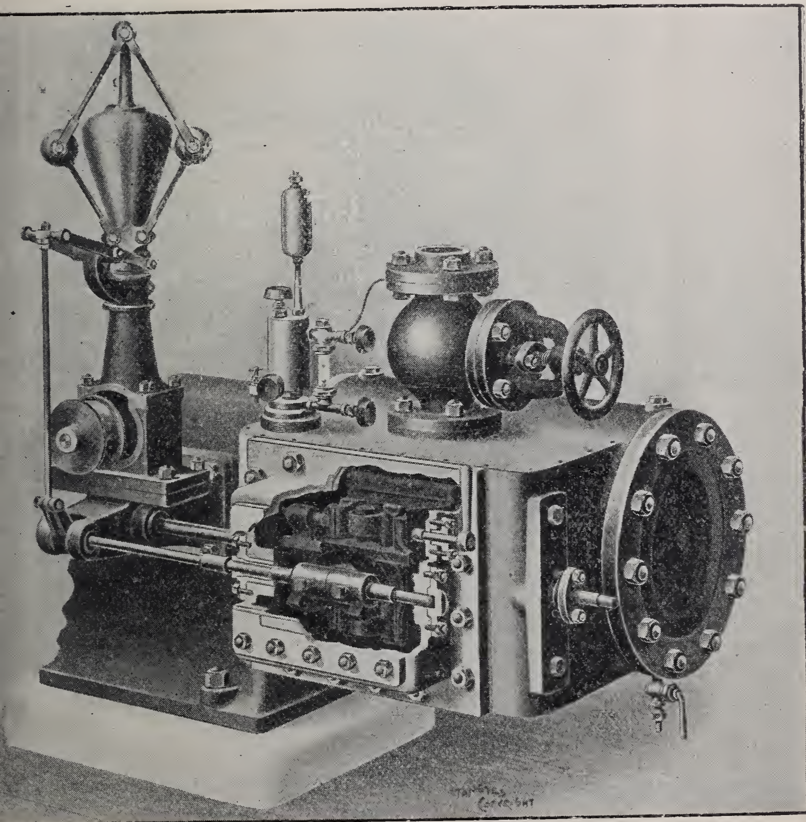


FIG. 48.

cylinder and then the other, as it comes in contact with one or other of a pair of right and left-hand cams fixed on a spindle lying in the direction of motion of the valve.

This spindle is connected, by a lever, with the governor, which, on rising or falling turns it, bringing the parts of the cams, which are nearer or further apart, into line with the rocker, causing it to close the port in the valve earlier or

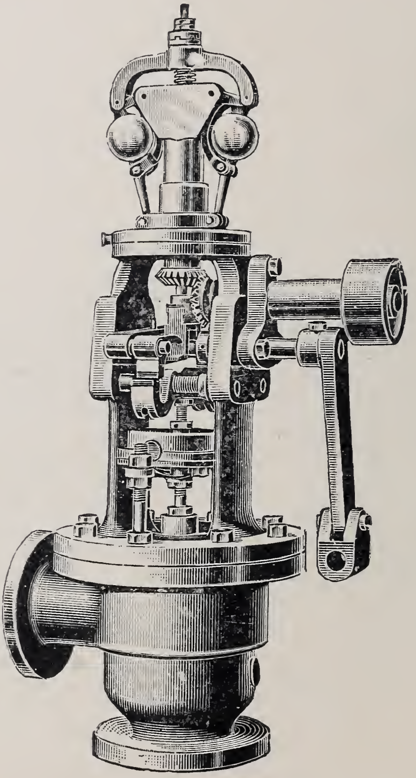


Fig. 49.

later, according to the time at which it comes in contact with the cams.

In this gear there is no sliding cut-off valve on the back of main to cause additional friction, or any second eccentric with its rods and link work, and yet it has the advantages

of the two-slide valve gear in that the exhaust is not interfered with.

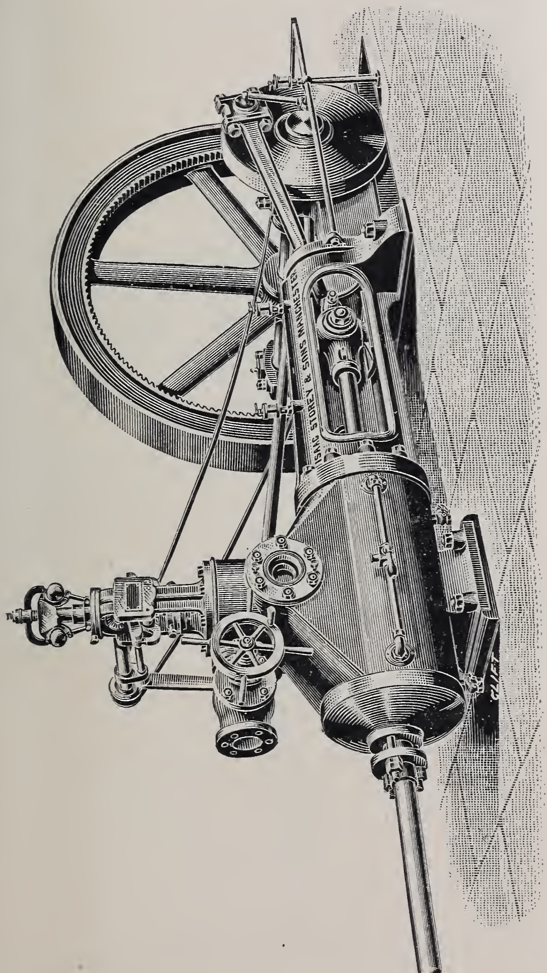


Fig. 50.

IV. *Proell Automatic Expansion Apparatus.*—This apparatus, manufactured by Messrs. Isaac Storey and Co.,

and illustrated by fig. 49, consists of a single drop valve controlled by Proell's patent governor.

The valve, which is an equilibrium valve, is opened twice during every revolution by the rock lever, shown in the

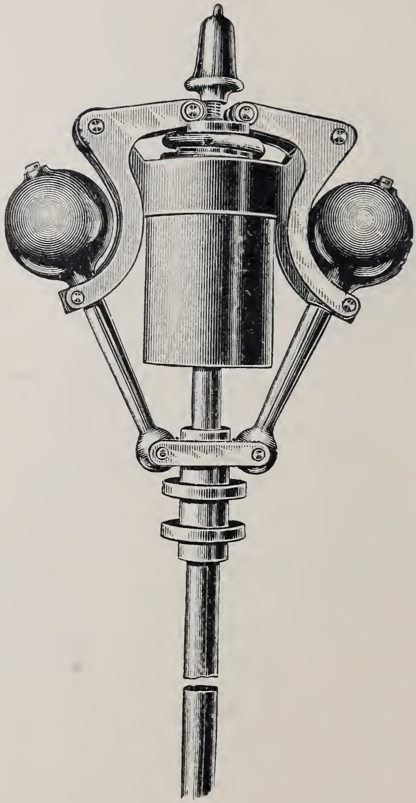


Fig. 51.

illustration, worked by an eccentric from the engine shaft, and is closed by a spring on being disengaged from the lifting lever, which disengagement occurs earlier or later in the stroke, depending on the height of the governor.

The apparatus is provided with the usual dash pot, which enables the valve to be closed without shock.

This apparatus is used in connection with an ordinary slide valve, which distrlbutes the steam to either end of the cylinder and also opens and closes the exhaust, or it may be used with a special form of Corliss distributing valve, worked by another eccentric, as shown in the illustration fig. 50.

Fig. 51 is an illustration of the governor used in connection with this apparatus.

This governor is arranged so that the balls move through equal distances with equal alterations of speed, also the arms of the governor are arranged in such a way that the balls, as they open out, remain in the same horizontal plane.

V. *Proell's Patent Two-Valve Releasing Gear.*—Fig. 52 shows this gear as made by Messrs. Marshall, Sons and Co. Limited, Gainsborough, and is described by them as follows :—

"The illustration represents a section of a cylinder fitted with Proell's patent two-valve gear. This gear consists of two equilibrium admission valves A, one to each end of the cylinder. These are alternately lifted by means of the valve spindle in connection with the principal trip levers B. These trip levers are depressed by means of the L levers C, which are connected on opposite ends of the rocking lever worked from the eccentric. The ends D of the L levers, projecting towards the centre, are brought into contact at each stroke with the trip pad E. This trip pad is connected with the governor through the medium of the levers F and G and the shaft H ; and the height of this trip pad determines the moment of cut-off of the steam, for as soon as the end D of the L lever reaches the trip pad any further movement of the rocking lever disengages the L lever from the trip lever B and the valve immediately closes. The closing speed of the valve can be very readily regulated by means of the spring box K, so that although it closes the valve very rapidly, it does it very quietly, thus preventing any undue wear of the valves or faces.

"All the working parts of this gear are case-hardened and the cutting-off edges are made of tool steel hardened.

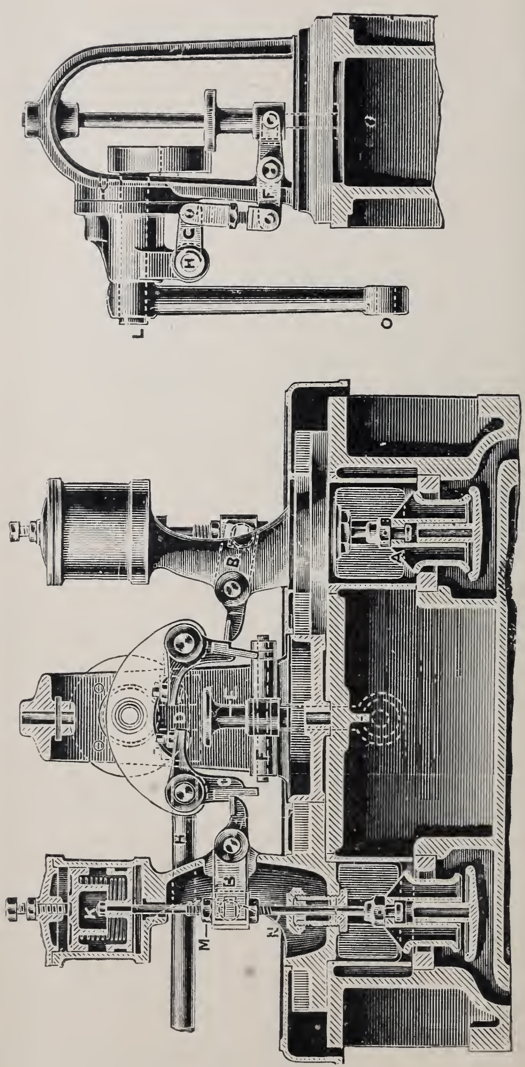


Fig. 52.

"The governor controlling this gear is of a very powerful and sensitive type, and has sufficient range to regulate the cut-off at any point up to five-eighths of the stroke. It can be fitted with a special adjusting arrangement, so that the speed of the engine may be varied 5 per cent either way whilst the engine is running ; this is a point of considerable importance for many classes of work, and enables the speed of the engine to be most accurately adjusted."

Fig. 53 shows Messrs. Marshalls' engine fitted with this gear and Corliss exhaust valves worked by a separate eccentric.

VI. *Richardson-Rowland Patent Drop Valve Gear.*—This is a two-valve releasing gear, but the valves are operated each by a separate small eccentric fixed on a side shaft, which lays parallel with the engine, and is driven at the same speed as the engine shaft by means of a pair of skew gears.

This expansion gear, illustrated by figs. 54 and 55, is made by Messrs. Robey and Co., Lincoln, and is described by them as follows :—

"A A, fig. 54, are the two admission valves, and being almost in equilibrium, are lifted with great ease by the small eccentric and rods worked from a shaft revolving at the same speed as the engine, and parallel with the bed-plate.

"The admission valve is raised at the commencement of the stroke and held wide open until the point of the cut-off is reached, when it is instantly relieved, and falls on its seat, an air cushion in the guide cylinder above preventing its falling too heavily.

"While the valves are alternately raised by the engine, the point at which they are dropped depends upon the position of the governor, which regulates the cut-off at any point from nothing to $\frac{5}{8}$ of the stroke.

"B B show the exhaust valves. These are triple-ported so as to give a wide opening with a small travel, and work with a minimum of friction. Being placed directly under the cylinder, this latter is kept perfectly drained, and both admission and exhaust valve being close to cylinder body, the loss of steam in ports is avoided."

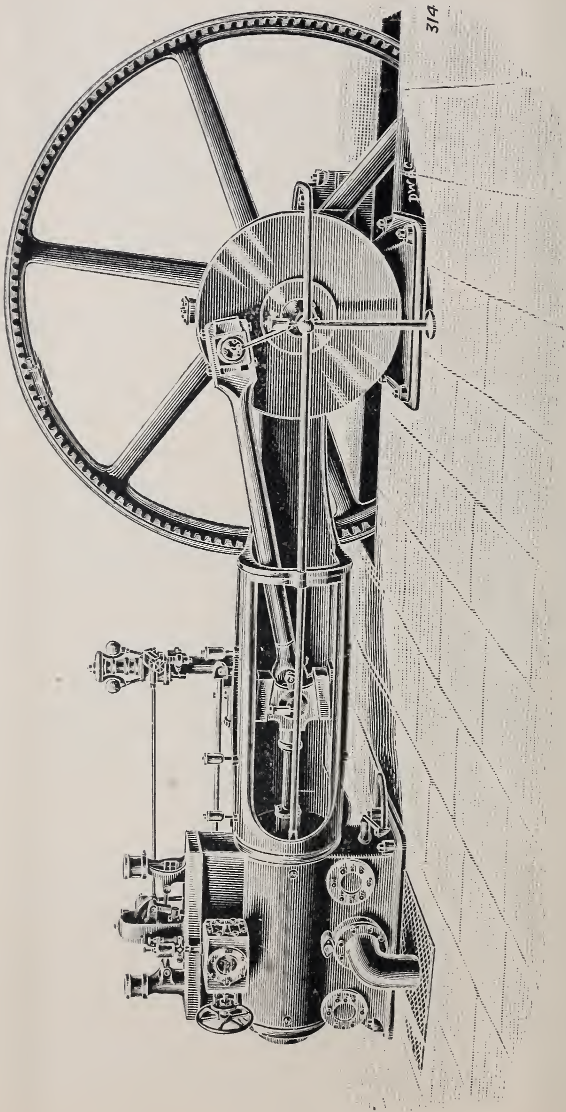


Fig. 53.

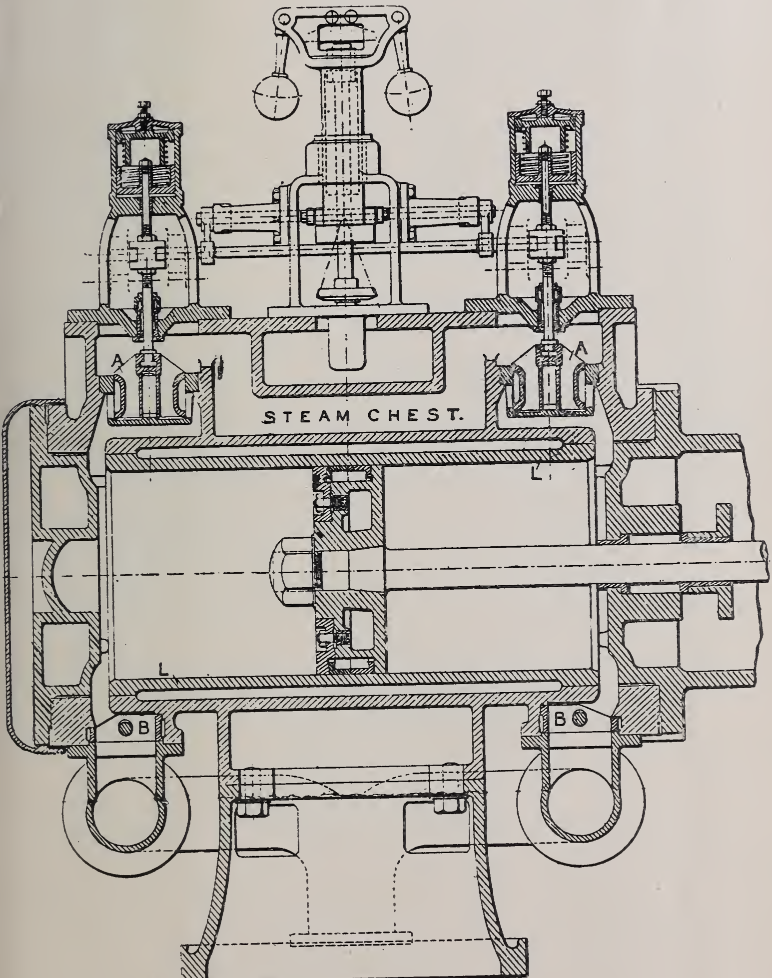


FIG. 54.

"Referring now to the cross-section of the cylinder, fig. 55 shows very clearly the whole of the gear for one end of the cylinder, it will be seen that the small eccentric rod K is enabled to act upon the valve A (which is in equilibrium) by depressing the outer end of the lever B, by which the valve is raised.

"This occurs just before the commencement of the stroke.

"Owing to the different arcs described by the end of the eccentric rod, and tbe lever B respectively, at a certain point the tripper L slips out of contact, and the valve drops instantaneously, cntting off the steam supply.

"The arrangement for securing an automatic cut-off is simplicity itself, the governor in rising moves the lever arm, and with it the pivot or fulcrum R of lever B, which, of course, has the effect of causing the tripper L to lose its contact, and the valve to fall at an earlier period in the stroke, and thus as the governor rises and falls the point o cut-off is accelerated or retarded accordingly.

"It will be noticed that the long horizontal arm of lever is prolonged past the governor, and that a small cord or string is attached to it. This cord may be led away to any part of the building, and forms a ready means of instantly stopping the engine in case of emergency, as in the case of accident to life or machinery. By pulling the cord, which is done with no more exertion than is required to ring an ordinary house bell, the lever G draws the valve-levers B completely clear of the trippers, when, of course, no steam can enter the cylinders, and the engine, which may have been exerting a thousand horse power, may be brought to a stand at once by the touch of a child.

"With so sensitive an apparatus as this, accurate governing is easy of attainment, and a small and simple centrifugal governor fulfils every requirement."

VII. *Frikart Corliss-Valve Trip or Releasing Gear.*—This gear, as manufactured until recently by Messrs. Greenwood and Batley, Leeds, is illustrated by fig. 56.

It is of the class in which the release can take place whether the valve is moving in an opening or closing direction, the trip catches having a positive movement

compulsorily bringing them back into engagement at the commencement of every stroke.

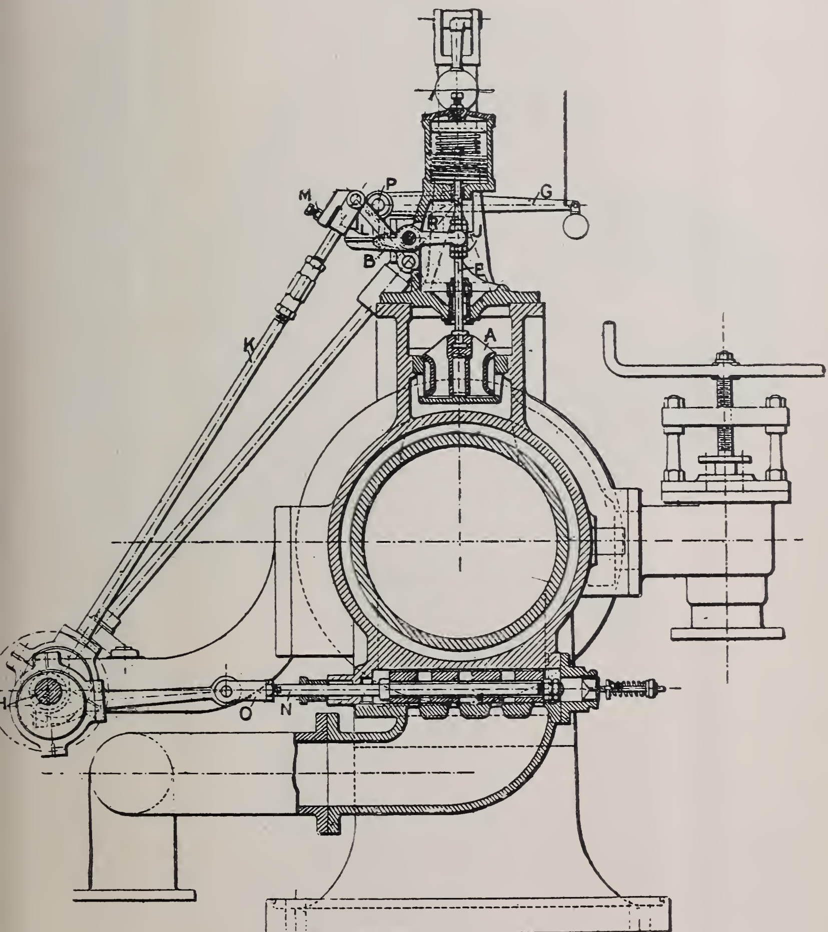


FIG. 55.

It is therefore possible to operate the steam valves from the same eccentric as the exhaust and yet have a late cut-off.

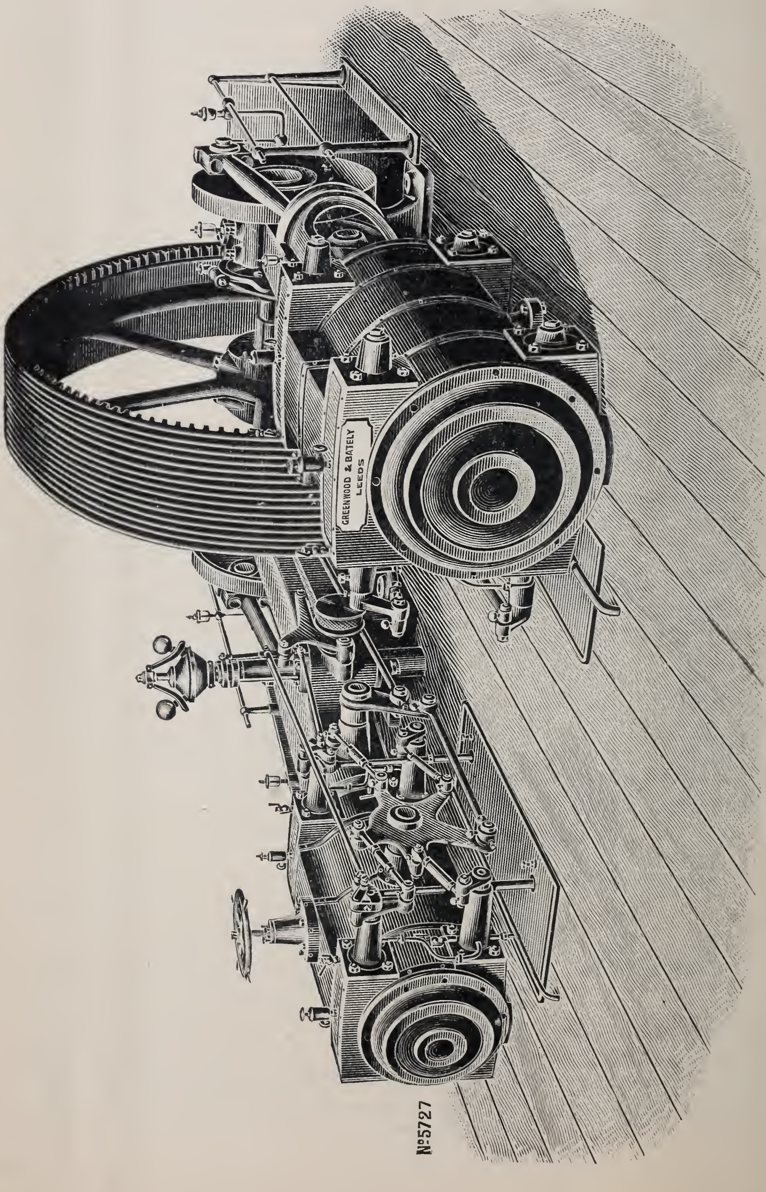


Fig. 56.

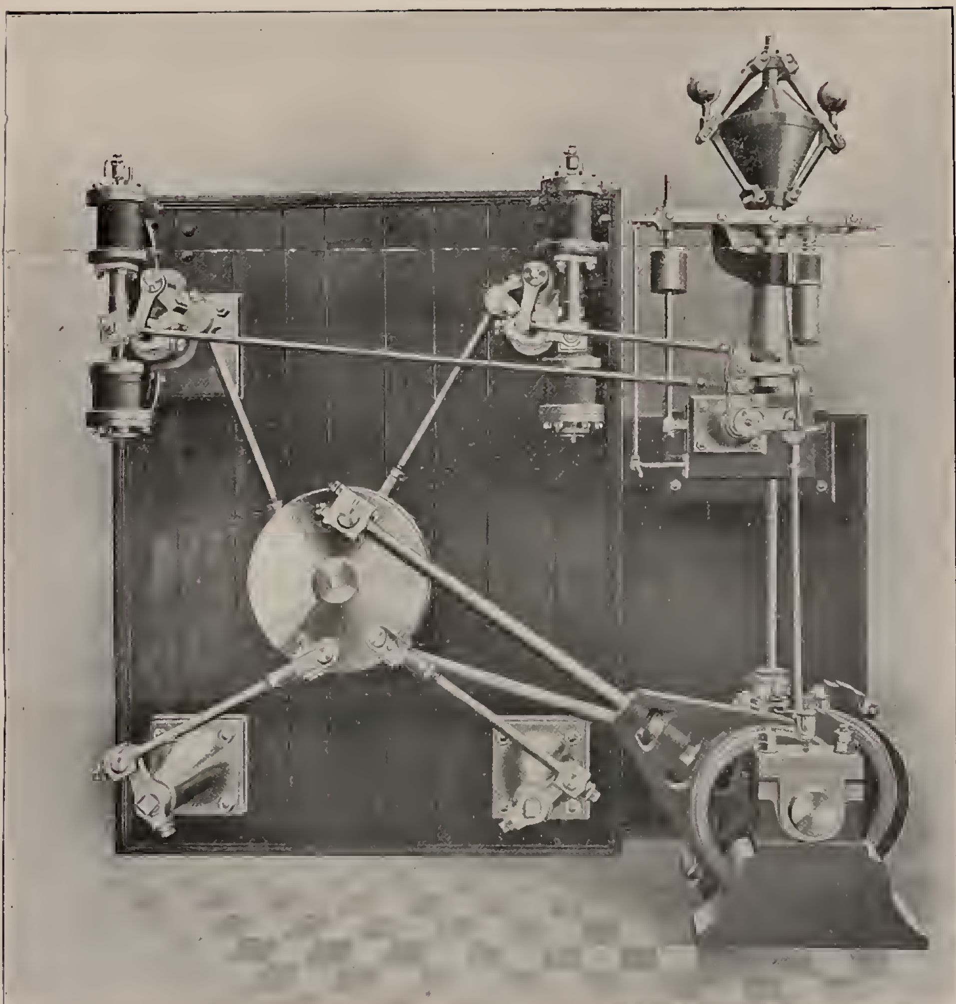


Fig. 58. GM

It will be noticed that the engine is fitted with a single eccentric and wrist plate for both steam and exhaust valves, the motion for the trip catches being derived from the vertical motion of the eccentric rod and taking place at an opposite time to the motion of the steam valve levers, the governor in rising or falling altering the path of the resultant compound motion so as to bring the catches out of engagement earlier or later in the stroke according to the "position" of the governor.

VIII. *Hick, Hargreaves' Patent Compensating Steam Dash-Pot for closing Corliss Admission Valves* is illustrated by figs. 57 and 58.

In describing this gear the makers say :—

"When the steam admission valves of a Corliss engine have been opened to the necessary extent by the positive action of the gear, they are disengaged by the trip motion controlled by the governor, and require to be closed promptly but without shock. The power required to effect this depends principally on the friction of the valves on their seats, which is proportional to the difference between the pressures above and below the valves. The means hitherto employed to provide this power have been either vacuum, pressure, or spring dash-pots of some definite strength without any provision for automatically adjusting their power to the resistances to be overcome. For factory engines where the minimum load, consisting of the friction of the engine and shafting, usually amounts to not less than 20 per cent of the full power of the engine, the pressure of steam in the cylinder during admission does not vary greatly over the whole range of load, and the power needed to close the valves being therefore practically constant, it may suitably be provided by dash-pots of the character described, sometimes needing assistance at or near the minimum load by throttling the steam supply at the stop valve. On the other hand, if the engine is required, as is usually the case with engines driving electric generators and some other classes of work, to deal with a load which may fluctuate over the whole range, from mere friction up to perhaps 50 per cent overload, and this with the full steam pressure constantly on the engine, such dash-pots as described are no

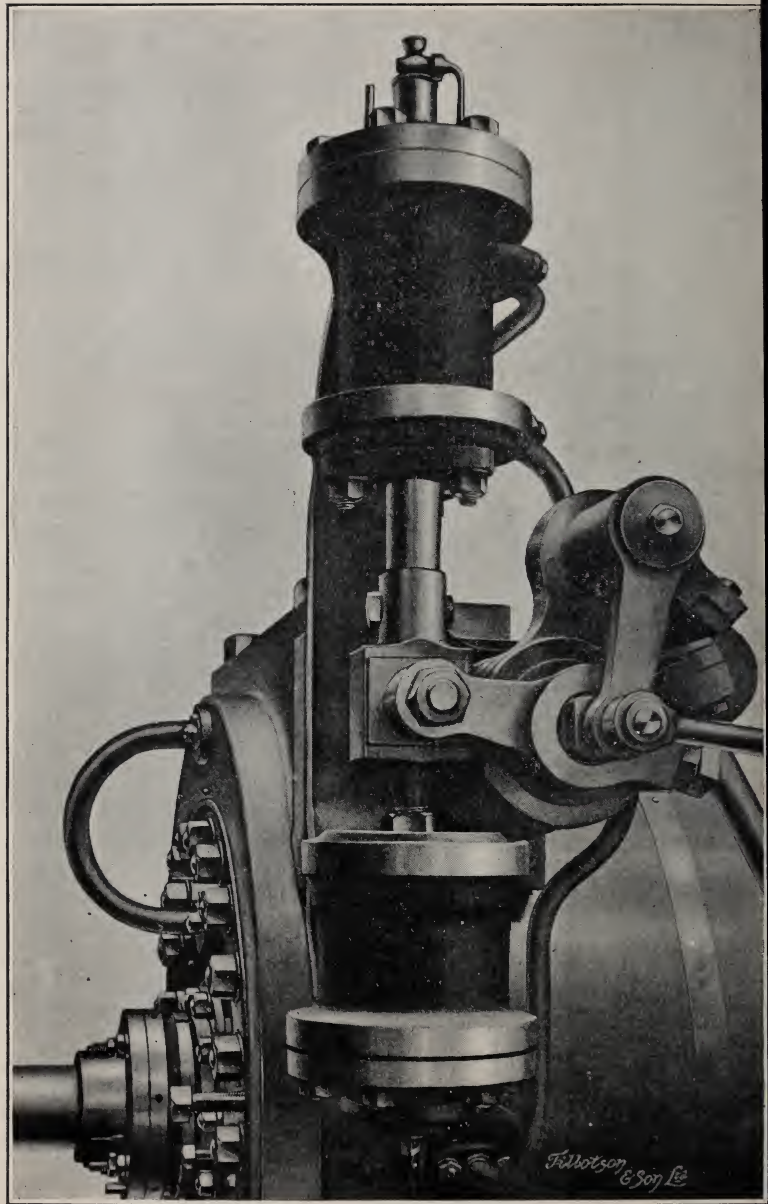


Fig. 57.

longer capable of meeting the requirements, as the trifling pressure of steam required in the cylinder to drive the engine at a very light load no longer approaches the pressure in the steam chest, and the great unbalanced pressure increases the friction between the valves and their seats to such an extent as to make the power required to move them four or five times as great as at moderate or full loads, and greater than the dash-pots can exert. The result is that the valves fail to close properly, and the engine runs irregularly and may even attain excessive speed if the gear is of a class which does not provide for the valves being positively closed on the return stroke, if not previously closed by the dash-pot. The defect described is common to all types of Corliss gear, and has caused some engine builders to employ drop valves, which, having no sliding friction, are free from this particular difficulty, though open to many other objections.

"The dash-pot shown in the illustrations is designed to overcome the difficulty, and does so automatically in the simplest possible way by employing the variable pressure which causes the varying friction of the valve to act also on a piston employed to close it, the result being that the power bears a constant ratio to the load. The dash-pot consists of an upper steam cylinder and a lower air cushion cylinder combined in one casting attached to the valve bonnet. The steam cylinder contains a simple piston, the upper side of which, like the admission valve, is exposed to the steam chest pressure, whilst its under side, again like the admission valve, is exposed to the cylinder pressure, the connections being made by the small copper pipes seen in the illustration. The piston rod of the small cylinder is made of a larger diameter than needed for mere strength, its area, acted on by the difference between the steam chest and atmospheric pressures, being made sufficient to overcome the constant friction and inertia of the parts, whilst the variable and larger proportion of the load is, to use the expressive American phrase, 'taken care of' by the varying pressures acting on the remaining annulus of the piston. The steam cylinder is proportioned to close the valve under its maximum coefficient of friction, and the air cushion cylinder

to take up any surplus energy, an air valve of special construction, capable of fine adjustment, being provided on its under side. This form of dash-pot, after more than a year's severe trial, has entirely fulfilled the makers' expectation, and has recently been fitted by them to five engines of from 1,450 to 1,600 I.H.P., one of which has been running for some time in a most satisfactory manner at the Leeds Corporation Tramway Power Station. It is from a photograph of this engine that the illustration has been prepared.

"The valve gear to which this dash-pot is applied is of the 'Frickart' type, in which the trip is actuated positively by a supplementary eccentric instead of by spring action and the opening motion of the valve, this securing several advantages, including an approach to positive action, allowing the valve gear to run at a high speed, and a full opening of the steam port at a much earlier point in the stroke than with Corliss gears of the more usual type. The gear, a complete set of which was exhibited at the Glasgow Exhibition, is made with very liberal proportions and surfaces, in view of the high speed and heavy loads for which it is designed."

IX. *Fraser and Chalmers' Reversing Corliss Engine Trip or Releasing Gear.*—Primarily this gear consists of the steam and exhaust valves driven from the same eccentric through an Allan link-motion reversing gear, the cut-off being effected by the lap of the valves. To this is added the Seymour patent auxiliary cut-off gear, controlled by the governor, and giving a range of cut-off extending from zero to 95 per cent of the stroke; the cut-off at both ends of the cylinder being even throughout the range, a feature not usual with ordinary gears.

The motion of this gear, which, as in the case of the "Frikart," takes place at an opposite time to the motion of the steam valve levers, is derived from a separate auxiliary eccentric, as shown in sketches, figs. 59 and 60.

A difficulty in applying any form of automatic governor gear to a reversing engine is as follows :—Should the engine suddenly be reversed when running at full speed, the governor being then in its top "position" and cutting off early in the stroke, very little or no steam will be admitted

to the cylinder to effect the reversal of the engine. The effect would be much the same as "back-pedalling" on a bicycle fitted with free wheel.

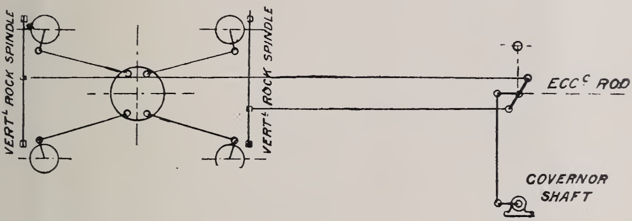


Fig. 59.

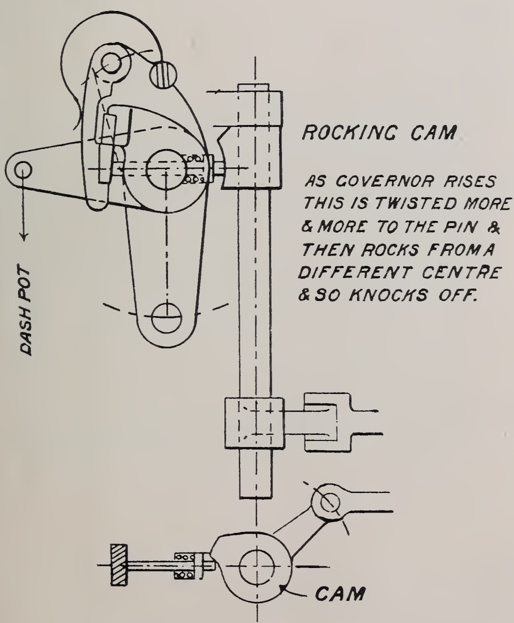


Fig. 60.

The patent Whitmore controller, a very ingenious arrangement of grab-hook, however, is fitted to the reversing

lever when this gear is used, by means of which, when the reversing lever is pulled over, a temporary engagement is made with a lever on the governor shaft, and the governor is

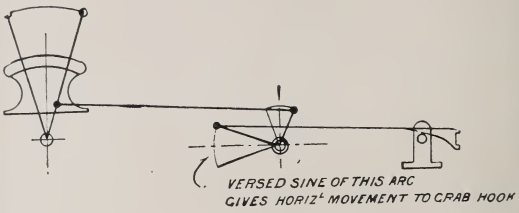


Fig. 61.

pulled down into its lowest "position," allowing the valves to admit full steam against the piston. This arrangement is shown in sketches, figs. 61 and 62.

The illustration, fig. 63, shows a Corliss winding engine built by Messrs. Fraser and Chalmers for the Sherwood Colliery Co., Mansfield, fitted as above. The governor is not seen in the illustration, being partly hidden behind the near cylinder.

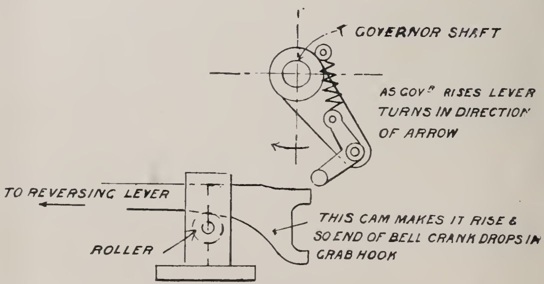
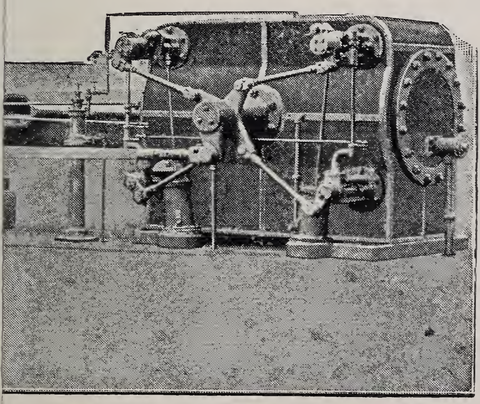


Fig. 62.

X. *Fraser and Chalmers' Combined Air and Speed Governor (Whitmore's Patent).*—This governor gear has been specially designed for air compressors and pumping engines.

"1. *Application to Compressors.*—The governing of a compressor does not appear to have had the attention it



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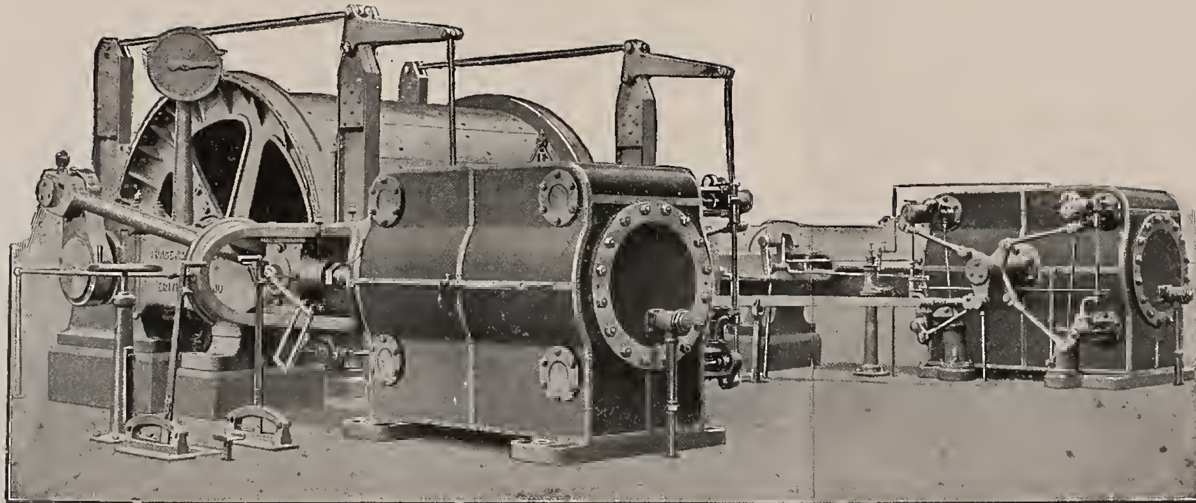

FIG. 63.

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warrants. It is apparent that it is useless to purchase an engine fitted with the best kind of valve gear and of first-class workmanship throughout in order to obtain the utmost economy, if the governing of the engine is at all erratic in its action, or if the engine is fitted without a governor at all, which is frequently the case in even very large compressors. There are, of course, various ways of governing the work of a compressor, but we are only considering the means that are most efficient. A compressor has to run under a good many changes of circumstances. First, regarding the steam engine: the steam is probably supplied to it from a battery of boilers which are also providing steam for a winding engine or other power which is very intermittent, and so extensive as to cause a frequent rapid reduction of pressure, which with some governing arrangements is quite sufficient to cause a disagreeable and wasteful hunting action, and what is still more annoying, the compressor will likely stop altogether if the air demand is small at the time of a sudden drop of boiler pressure. Of course, this stopping does not matter where the engine is of the wasteful cross compound or duplex type taking steam practically full stroke on both sides, as it will probably start away again as soon as more air is required. Where the engines are fitted with cut-off valves on both sides, the compressor will not start again when more air is required without first putting into or out of action some valve or other by hand, or barring the engine round by hand. It is then that stopping is a great nuisance.

"There is another point in considering the governing of compressors fitted with Corliss or drop valves which frequently has been the cause of considerable hunting in speed—that is, the difference in the point of cut-off when running at a high speed and when running very slowly. The higher the maximum piston speed of the engine, the more pronounced will be this difference.

"Considering a compressor which varies in speed according to the amount of air required, and where the point of cut-off is controlled by the air pressure, it may be at any time from minimum to maximum capacity. The mean pressure is practically constant whether running 20 or 70. The only difference in the steam pressure would be due to the higher

velocity of steam and air through ports, which is trifling. In a trip gear engine, the point of cut-off is, however, not constant, as the closing of the steam valves after they are tripped takes time. This being the case, the point of cut-off must be later when running at a slow speed than when running at a high speed, in order to produce the same mean pressure. This action tends to make the engines hunt when controlled by certain designs of governors. Suppose a compressor fitted with one of these governors is running at 20, and the tripping point in the steam cylinder is at 25 per cent of the stroke, the actual cut-off point would be almost directly after, say 26 per cent. In other words, there would be a sharp corner in the diagram owing to the compressor running slowly. Now, suppose a much greater volume of air is required, the pressure will begin to drop in the air receiver. This produces a later cut-off, say 27 per cent. The compressor at once increases in speed and is still further accelerated in its speed ; secondly, by the drop of pressure in the air receiver ; thirdly, by a considerably increased cut-off due to the piston speed being greater, whereas the steam valves take just as long to close as when running slowly and so adding 2 or 3 per cent to the 27 per cent cut-off. The result is the combined forces drive the engine to a higher speed than necessary and so introduces a hunting action.

"Now, regarding the air pressure, the engine must not be allowed to run away if the air mains are opened up as to reduce the air pressure to only a few pounds, the pressure should not be allowed to vary more than 2 or 3 lbs. per square inch at no matter what demand, provided it is within the limit of the compressors. The governor should be such as to control the engine at the slowest possible speed, at which time the air that is delivered will escape through receiver relief valves (which would be set a few pounds above ordinary pressure) when there is no demand for air, and this irrespective of any sudden drop in boiler pressure. The efficient governing of a compressor presents a point of great economy in a high-speed compressor. It is here that one of the advantages of a high-speed compressor, such as the Riedler, is apparent. A Riedler compressor may be run

at 100 revolutions per minute, compress a certain amount of air, and run at a speed as low as 8 revolutions per minute to compress a minimum amount of air. Now compare this with a compressor, the maximum speed of which is only 40 revolutions per minute and cannot be run slower than 8 revolutions per minute without stopping. When no air is required, which happens frequently during the day, the Riedler will only be blowing away 8 per cent of its maximum capacity, whereas the other compressor will be blowing away 20 per cent of its maximum capacity. Many compressors at mines, collieries and elsewhere, can be seen running daily at the maximum speed that can be got out of them, or at least a proportionately high speed, and probably only one-half of the air is used, the other half is blowing into the atmosphere, consequently half the coal, men's time and boiler wear, tear and capacity, is being wasted also.

"To reduce this excessive waste, a number of air receivers are often installed, representing a great volume of stored compressed air, which will certainly provide for any slight fluctuation in the air demand, but it is surely an expensive means of partly overcoming the difficulty of wasted compressed air when efficient governing and one comparatively small receiver will entirely overcome the trouble. In fact, the air mains down the shaft have been found to be ample receiver space to enable the compressor, properly controlled, to do this work without waste of air without the aid of any receiver on the surface at all."

Construction.—Seeing the importance of compressor governing, and the great saving in fuel, etc., that is effected thereby, Messrs. Fraser and Chalmers have now introduced a combined air and speed governor which fulfils all the conditions required.

"This governor is shown in fig. 64 as usually arranged, and figs. 1, 2, and 3, diagram 1, show section through same. The apparatus consists of two main portions, a centrifugal speed governor A and a compressor governor B. This latter consists of a cylinder D, having a plunger C connected to a spring V secured to the bottom of the cylinder, where screw for slight adjustment is arranged. The upper part of the plunger is connected by necessary levers and links to one

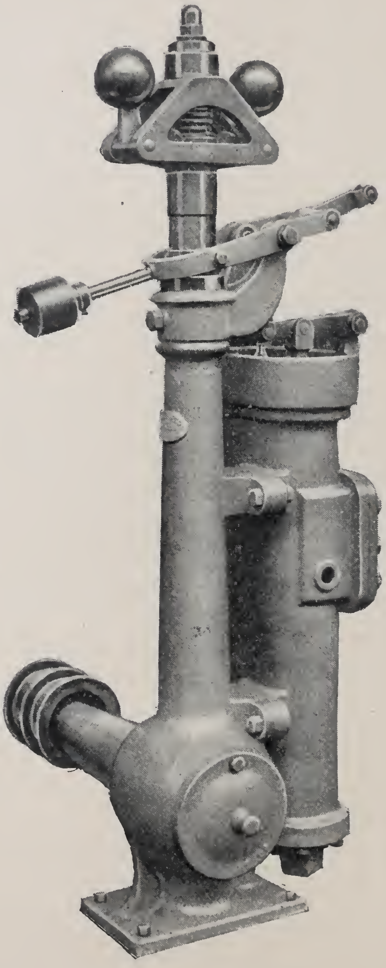

FIG. 64.

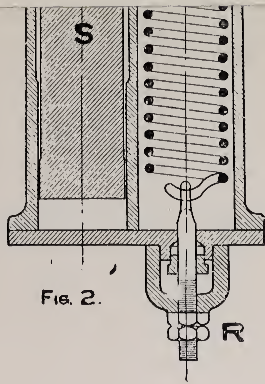

Fig. 2.

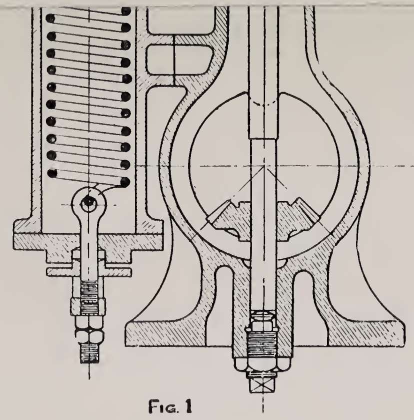

Fig. 1

Diagram 1.

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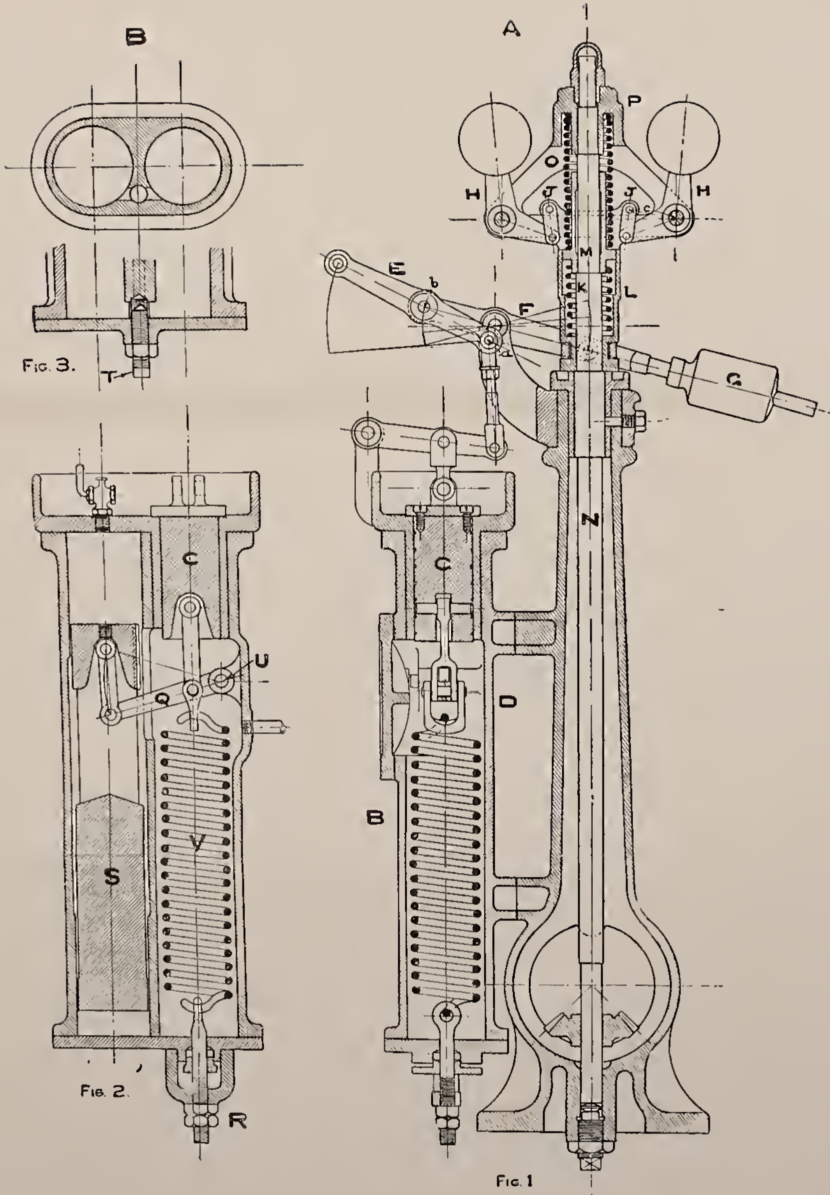

DIAGRAM 1.

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end of floating lever E. This floating lever is pivoted on the bridle lever F, which is in connection with fly-ball governor sleeve. The outer end of this floating lever E being adapted to be attached direct or by suitable connections to the cut-off or switch mechanism, so that any movement of this end will regulate the speed of the engine or motor.

"In addition to the plunger C there is an internal weight S connected by certain levers to the plunger C, so arranged that the travel of the weight is greatly multiplied over the travel of the plunger. The flow of oil from one end of the weight to the other is regulated from outside the cylinder by an adjusting screw T (fig. 3, diagram 1), so that the speed or movement of the pressure plunger C can be controlled as desired.

"As the weight has a considerable movement, the passage through the bye-pass at screw T is not so small as to become clogged.

"The speed governor A consists of a governor head P carrying the fly-balls. These are connected by links (the special angle of which levers and links play an important part in this governor) to a spring box L, which is connected by a loose collar to the bridle lever F.

"In this spring box is contained a short spring K, and in addition to this there is an independent sleeve M bearing upon a shoulder on the governor spindle N and between which and the governor head P there is a long spring O fitted.

"The usual design of speed and air governor consists of a speed governor of fly-ball type which is loaded either with a spring or centre weight, which spring or weight comes into action only when the compressor exceeds a certain speed near to the maximum speed of the engine, say three revolutions under the maximum speed. This governor so has nothing to do with the controlling of the compressor at any speed under the same. Since the compressor is working practically all the time under the maximum speed, the speed governor has therefore nothing to do with the controlling of the engine excepting in case of accident or any case where maximum revolutions is obtained. The air

governor has therefore all the controlling to do; this consists of a spring-loaded plunger, the air acting upon the under side of same, forcing out the plunger, which action would reduce the cut-off according to the amount forced out.

"If the quantity of air to be supplied were constant, and the steam pressure constant, this arrangement would answer very well, but in no case can either constant steam pressure or constant quantity of air be relied on, and for this reason this style of governor cannot be called satisfactory.

"As already described, the fly-ball governor is fitted with two springs, K O; these springs come into action at different periods. Considering first spring K, it is of such a free length that when the governor sleeve L is about $\frac{3}{8}$ in. from its lowest position it is just holding the spring against its upper bearing—that is, the sleeve L can be lifted $\frac{3}{8}$ in. without compressing any spring; a small weight, or even the governor levers themselves, being sufficient to bring the governor to its lowest position when standing or running very slowly.

"The spring K is of such a strength that the governor will gradually rise as the speed increases until it becomes close coiled or its upper and lower bearings are by other means brought into contact with each other, at which time the compressor will be running at its so-called maximum speed, the sleeve L will be, say, $2\frac{3}{8}$ in. above its lowest position—*i.e.*, for every different engine speed there is a corresponding position of governor sleeve.

"This is one of the important features of the governor—the entire absence of engine hunting being due to this feature, as it will be seen by reasoning out the action of the combined governor that the above-mentioned accelerating and retarding engine forces which create a hunting effect are quickly curtailed. Suppose the compressor running at, say, half speed, the air plunger C would be about half of its total travel up or outwards, the fly-ball governor sleeve about $1\frac{3}{8}$ in. above lowest position. The air will be at normal pressure. Now suppose there is a greater demand for air, the air pressure will begin to drop and very soon the plunger C will be drawn inwards. (It may be mentioned here that the plunger is usually made to travel its full

distance with a total rise or fall of pressure of 5 lb. per square inch, or about 7 per cent of the required air pressure.) With the altered position of plunger C there is an extended cut-off given to the steam cylinder gear. Now, if the fly-ball governor could not change its position, hunting would probably commence, due to this and the other force mentioned, but in this case the fact of the engine increasing in speed is the means of again shortening the cut-off and so preventing hunting.

"The $\frac{3}{8}$ in. free movement of governor sleeve L without spring pressure is introduced to prevent engine stopping altogether, the weight controlling this part being so adjusted as to pull down the governor to its lowest position when running at its minimum speed, and so considerably increasing the cut-off for one stroke when just on the verge of stopping. The upper spring O only comes into action when the compressor exceeds its maximum speed by, say, 5 per cent, then, as this is a very long spring and loaded to a safe limit, it will yield quickly, *i.e.*, it will allow the sleeve L to rise another inch by the increase of speed of about five revolutions per minute. This spring comes into action when the demand for air exceeds the maximum limit of the engine. In fact, the various levers are so proportioned that the cut-off can be so shortened as to prevent the engine exceeding 5 per cent above its maximum speed when the air cylinders are doing no work at all.

"As an explanation, suppose the gear that drives the speed governor were to fail, the lever F would drop to its lowest position and remain there. The gas or air pressure rising, due to the increased speed of the machine, would force the plunger up and with it the inner arm of the floating lever, thus reducing the supply of steam and preventing the machine from running away. Or again, suppose a greater amount of air is being drawn off than the compressor is capable of supplying, causing the gas or air pressure to drop, the plunger would then drop to its lowest position and remain stationary, the speed of the machine would then be controlled entirely by the speed governor.

"The advantage gained by having the arrangement of two separate springs as described above can probably be best

understood from an actual example. Take the case of an air compressor working as described above, *i.e.*, its maximum working speed of 75 revolutions per minute. The speed governor will then be set so that at this speed the internal shoulder on the lower spring box L shall be just touching the collar on the spring sleeve M so that for any speed less than 75 revolutions per minute the balls will be acting against the lower spring only. Now, supposing a greater amount of air is being drawn off than the compressor is capable of supplying, causing the gas or air pressure to drop, the compressor would want to race away, but by the time the speed had increased to about 80 revolutions per minute the governor would have risen to its highest position against the resistance of the upper spring O, thus shutting off the supply of steam and maintaining a safe speed.

"*Advantages.*—In short, the result of this special design of speed governor is that a steady speed is maintained according to any amount of air required between minimum and maximum. In other words, the speed varies exactly as the volume of air required varies. The difficulty of hunting that has been apparent in other designs of combined speed and air governors is entirely overcome, and it is not dependent on its driving gear nor the pressure being maintained for the safety of the compressor.

"*Applied to Pumps.*—Precisely the same governor is employed for pumping engines, the pressure governor being connected to an air vessel in the rising main, and is specially suitable for pumping engines that are pumping against a constant head and which require to run according to the amount of water that is drawn away. For instance, in waterworks' stations where the quantity of water varies from maximum to minimum during the day, and where a certain pressure should be maintained all the time, the governor acts to perfection when controlling such a pumping engine.

"*Applied to Constant Load but Varying Speed Engines.*—Where a pumping engine or compressor, or any engine that is working under a constant load, but where it may be required to vary the speed from time to time, the Whitmore governor is a very suitable arrangement. In this instance

the pressure, or plunger part of the governor is not required. The cut-off switch or throttle of the engine or motor is connected direct to the outer end of the bridle lever F, and a hand wheel adjustment is provided in the rod for lengthening or shortening the rod which connects to the bridle lever. By this means the engine may be run at any speed that is desired between minimum and maximum by the adjustment of this hand wheel, and should the load be suddenly released from the engine, no matter at what speed the engine is adjusted for, the engine will not exceed its maximum speed. Special arrangement is made in this case so that should the governor belts break, the stopping of the governor would shut off steam entirely, so that this governor has the advantage of absolute safety when employed in circumstances of this kind, as well as in the case of pumping engine and air compressors as before-mentioned."

XI. *Tangye's Shaft Governor.*—Fig. 65 shows Messrs. Tangye's improved shaft governor. The following is a list of parts:—A, fly wheel; B, inertia arm, carrying eccentric; C, eccentric; D, centrifugal weight box; E, tension spring; F, tension screw; G, tension spring box spanner; H, inertia arm pivot; I, centrifugal weight box pivot; J, spring shackle and pin; K, connecting link; L, eccentric strap; M, eccentric rod and joint; N N, stops; O, lock nuts.

This governor consists of one centrifugal weight D, one spring E, and the inertia arm B, carrying eccentric C, which is directly connected by rod and joint M to an equilibrium piston valve.

The centrifugal weight is pivoted at I, and is connected to the inertia arm by link K.

The inertia arm carrying eccentric is pivoted at H, and when the speed is suddenly increased it swings the eccentric across the shaft from position of maximum to that of minimum travel, at which the cut-off is nil; the centrifugal weight also at the same time flying out and maintaining eccentric in position required for the altered load.

This governor is very simple and efficient, and is used by Messrs. Tangye for controlling the speed of both steam and gas engines.

XII. *Robinson's Patent Shaft Governor.*—This governor is shown in fig. 66. Its special features are :—

1. The eccentric is supported and guided by an arrangement of jointless spring connections.

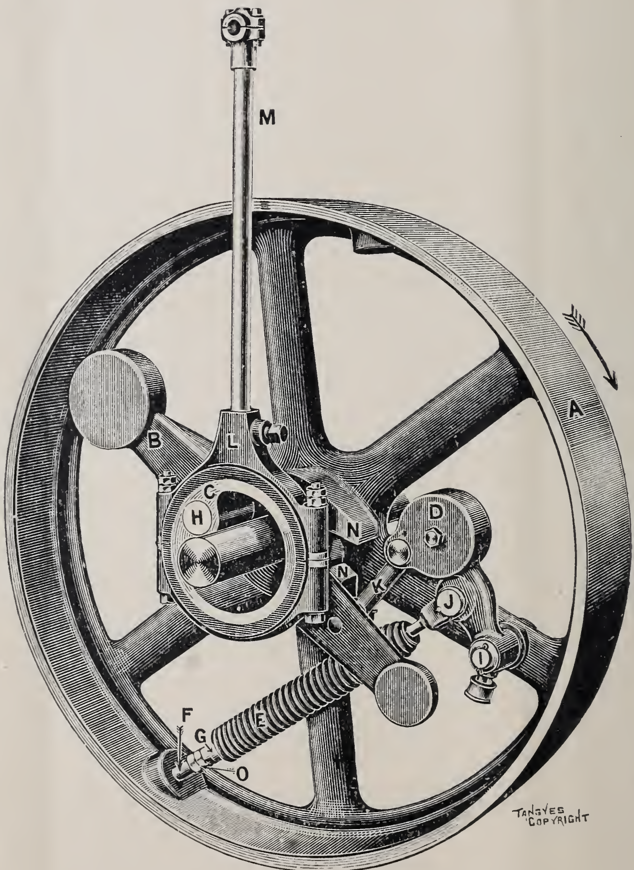
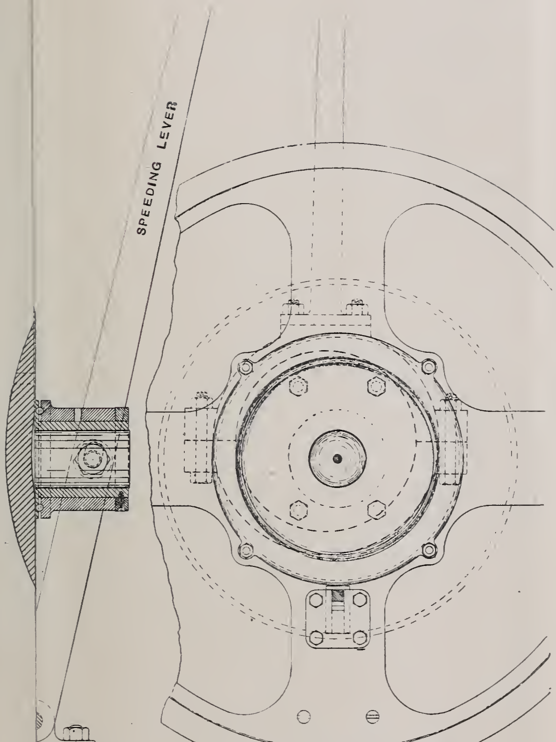


FIG. 65.

2. The governor has no spiral spring or springs, but the centrifugal force is balanced by the resistance to bending of



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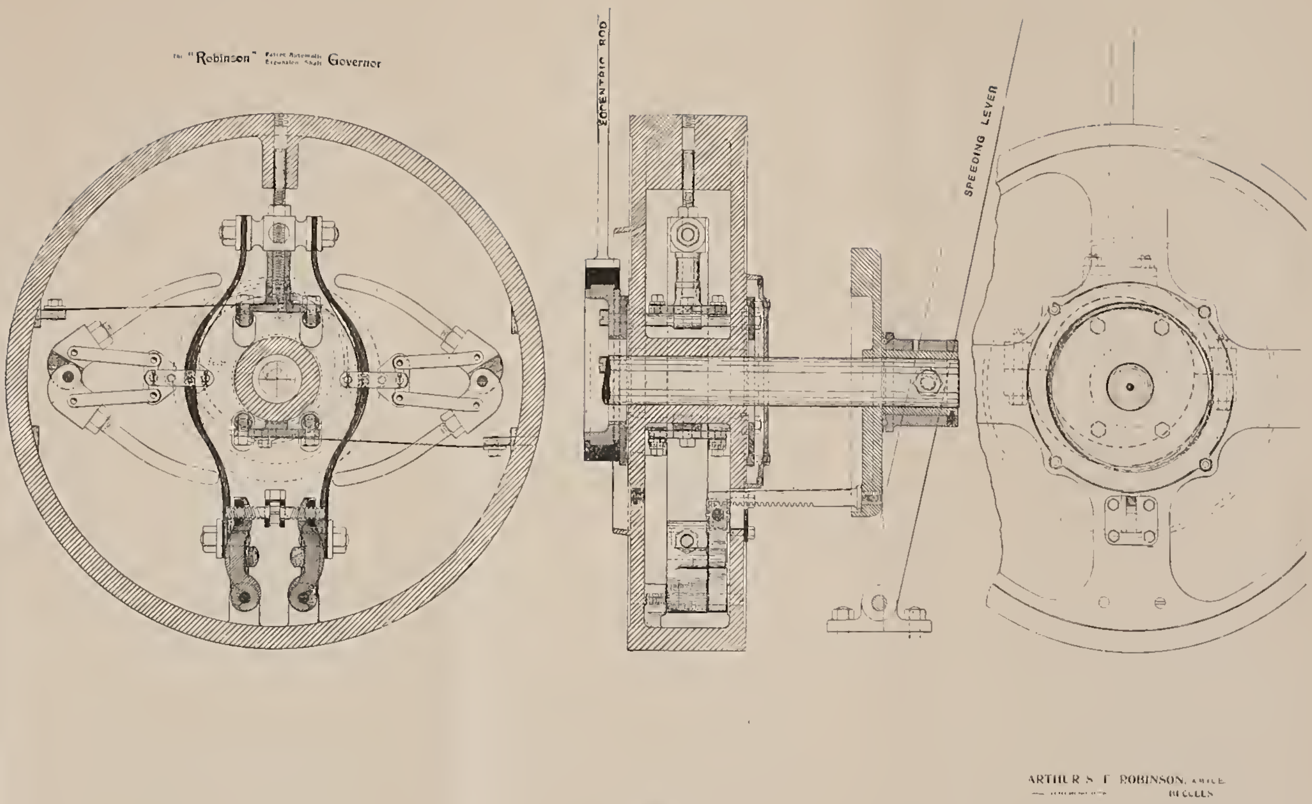
The "Robinson" Patent Automatic Expansion Shaft Governor
ECCENTRIC ROD
SPEEDING LEVER
ARTHUR S. T. ROBINSON, A.M.I.C.E.
BECCLES

Fig 66.

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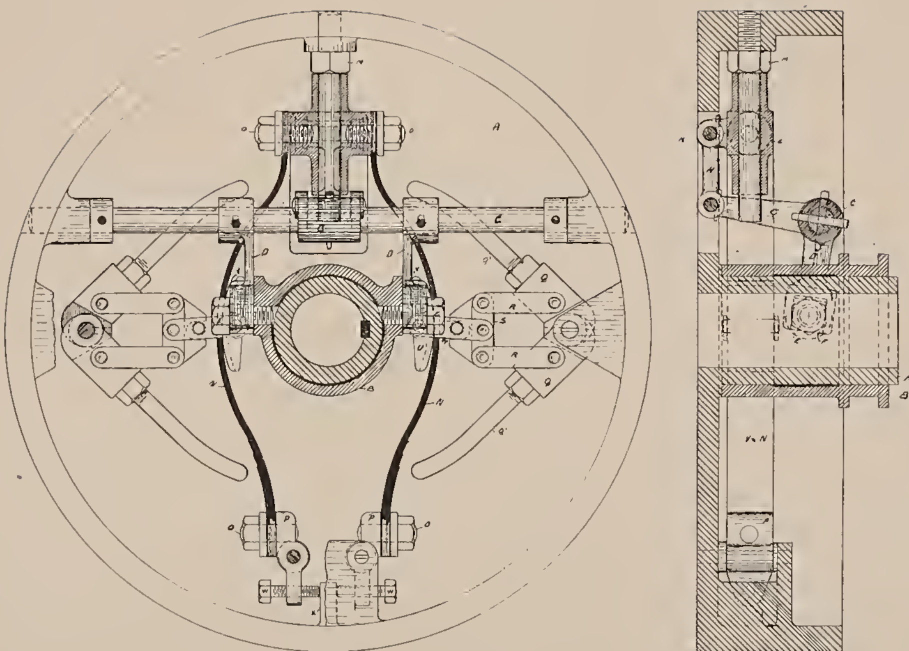

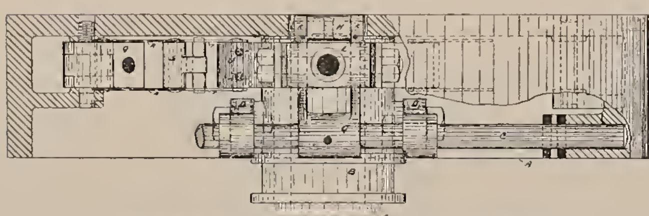

DESCRIPTION

A – Governor wheel & boss –
B – Sleeve – a sliding fit on A –
C – Lever shaft – carried by wheel –
D – Sleeve levers – fixed to C –
E-F – Screw & block – connecting D D to B –
G – Crosshead lever – fixed on C at at angle to D
H – Link – connecting G to crosshead lugs K –
L – Crosshead – a sliding fit on stud M –
N – Laminated steel spring blades –
O – Studs – securing blades N to L & hinges P –
Q – Lever arm joints, fulcrumed to wheel and connected to shoe U by links R – Compensator S and links T ——

The shoe U is held in place by the blade V which is fixed at one end to the shoe and at the other end with the spring blades at O –

Action of Governor

The levers Q Q' move outwards under centri-fugal force; when this is sufficient to overcome the resistance of the springs N N; drawing the crosshead L towards centre of wheel, which movement is imparted to the shaft C & sleeve B through the link H & levers G & D – The resistance of the springs is increased by adjusting the set screws W against the lug X so that the hinges P are moved nearer together – This means of varying the speed at which the Governor will control the engine when set in motion, may be applied to this in the same manner as in the Robinson Patent Expansion Governor ——

THE ROBINSON PATENT GOVERNOR
AS ARRANGED FOR CONTROLLING THROTTLING AND AUTOMATIC CUT-OFF GEARS

ARTHUR S. F. ROBINSON, A.M.I.C.E.
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FIG. 67.

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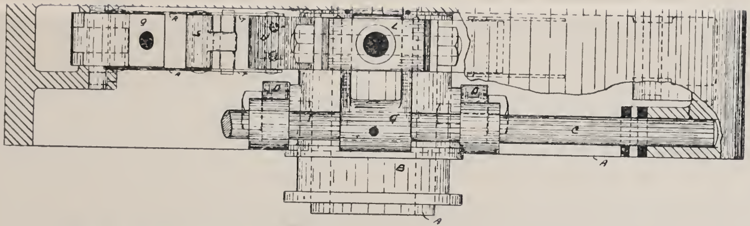

C – LEVER SHAFT – CARRIED BY WHEEL –
D – SLEEVE LEVERS – FIXED TO C –
E–F – SCREW & BLOCK – CONNECTING D D TO B –
G – CROSSHEAD LEVER – FIXED ON C AT RT ANGLE TO Q
H – LINK – CONNECTING G TO CROSSHEAD LUGS K –
L – CROSSHEAD – A SLIDING FIT ON STUD M –
N – LAMINATED STEEL SPRING BLADES –
O – STUDS – SECURING BLADES N TO L & HINGES P –
Q – LEVER ARM JOINTS – FULCRUMED TO WHEEL AND CONNECTED TO SHOE U BY LINKS R – COMPENSATOR S AND LINKS T

THE SHOE U IS HELD IN PLACE BY THE BLADE V WHICH IS FIXED AT ONE END TO THE SHOE AND AT THE OTHER END WITH THE SPRING BLADES AT O –

– ACTION OF GOVERNOR –

THE LEVERS Q Q' MOVE OUTWARDS UNDER CENTRIFUGAL FORCE; – WHEN THIS IS SUFFICIENT TO OVERCOME THE RESISTANCE OF THE SPRINGS N N; DRAWING THE CROSSHEAD L TOWARDS CENTRE OF WHEEL, WHICH MOVEMENT IS IMPARTED TO THE SHAFT C & SLEEVE B THROUGH THE LINK H & LEVERS G & D – THE RESISTANCE OF THE SPRINGS IS INCREASED BY ADJUSTING THE SET SCREWS W AGAINST THE LUG X SO THAT THE HINGES P are MOVED NEARER TOGETHER – THIS MEANS OF VARYING THE SPEED AT WHICH THE GOVERNOR WILL CONTROL THE ENGINE WHILST IN MOTION, MAY BE APPLIED TO THIS IN THE SAME MANNER AS IN THE ROBINSON PATENT EXPANSION GOVERNOR

THE ROBINSON PATENT GOVERNOR
AS ARRANGED FOR CONTROLLING – THROTTLING – AND – AUTOMATIC – CUT-OFF – GEARS –

ARTHUR S. F. ROBINSON, A.M.I.C.E
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FIG. 67.

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flat laminated steel blades so arranged as to form a spring, the strength of which can be adjusted whilst the engine is in motion and the speed of the engine varied as desired.

3. All the parts are in tension, and the lever arms, which constitute the "balls" of this governor, balance one another and are arranged in such a way as to be free from inertia forces due to change of speed.

Fig. 67 shows this governor arranged to operate a throttle valve, the motion for the throttle valve being taken off the sliding sleeve.

XIII. *Butler's Fly Wheel Governor for Gas and Oil Engines*, manufactured by Messrs. Clark Chapman and Co.

Fig. 68 shows this governor, which operates a throttle valve on the "mixture" supply, and is described as follows:—

"The governor consists of two wings, G^1, G^2, pivoted on the arms of the fly wheel W. Centrifugal force is resisted by the helical springs S^1, S^2, the ends of which rest against stops P, P. Each wing has a portion of a thread T, which actuates the sleeve E through the rollers R^1, R^2. The sleeve is prevented from rotating by the rollers R^3, R^4, running on the guide rails L. The endways movement of the sleeve is communicated to the governor shaft A by the fork F. The two wings are held in unison by the links D^1, D^2. This governor is very powerful, and being actuated by inertia as well as centrifugally, the pulsating action of the engine shaft is sufficient to move it with the least change in the engine speed."

For the 100 I.H.P. 3-cylinder compound gas engine made by this firm, which governs within 5 per cent from full load to no load, and does not hunt, the governor is mounted on a separate disc independent of the fly-wheel.

Fig. 69 shows the throttle regulator used in connection with the above governor on Messrs. Clark Chapman's gas and oil engines, which have always been governed by "change of volume," and consequently by varying the compression—the system believed by the makers to be generally recognised as the most suitable for large power engines and even for car motors.

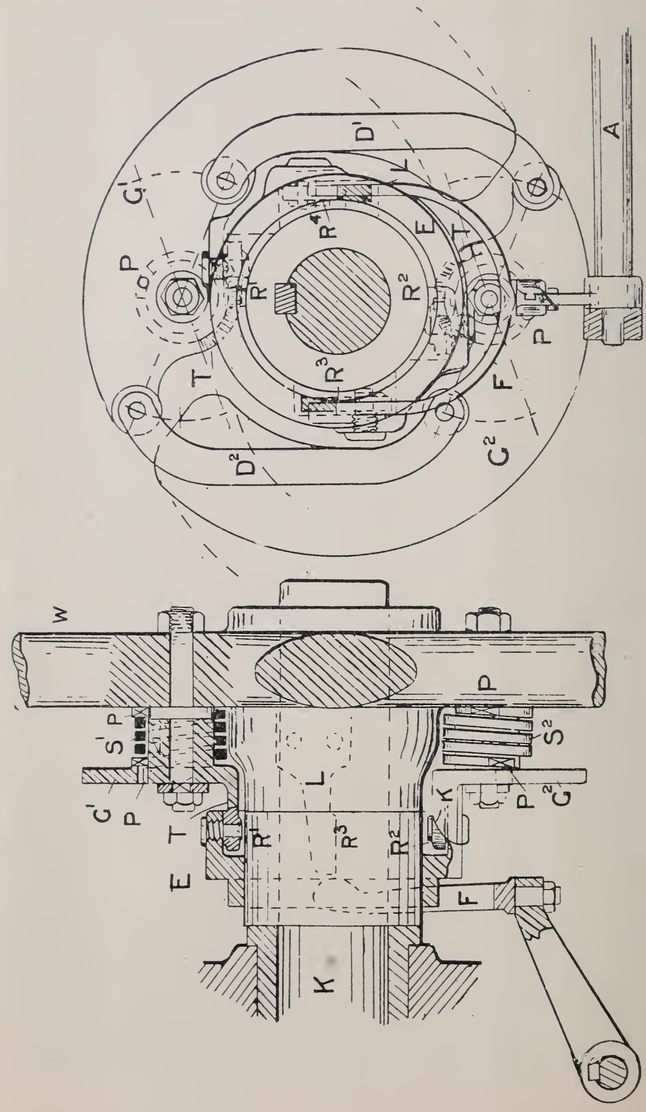


FIG. 68.

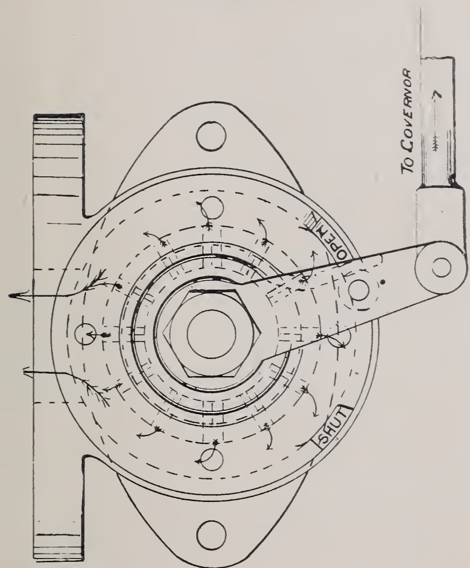
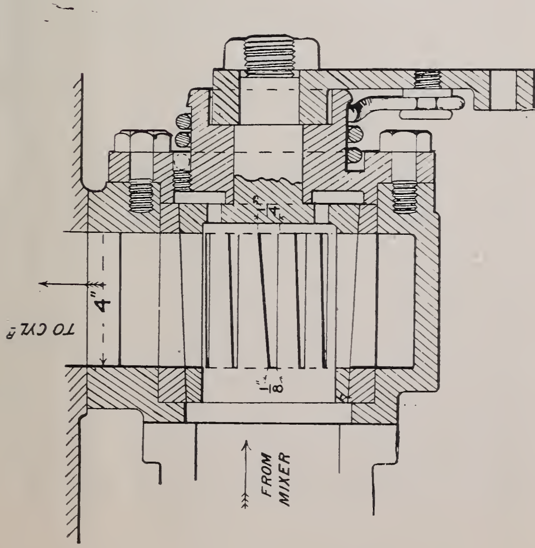


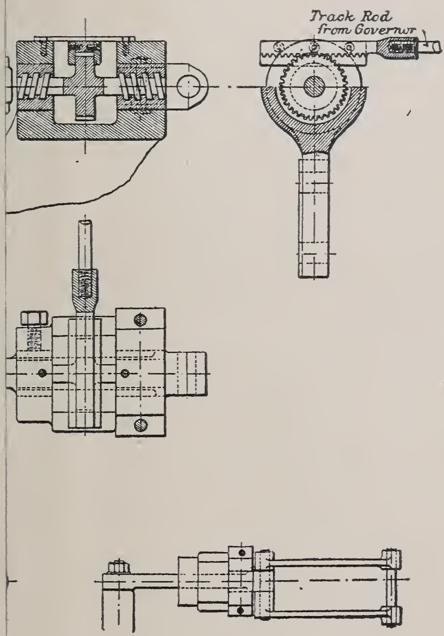
Fig. 69.



XIV. *Crossleys' Automatic Cut-off Gear for Gas Engines.*—In his paper on "Recent Progress in Large Gas Engines," read before the British Association at Belfast, September 11th, 1902, Mr. Humphreys speaks of this gear as follows :—

"The great regularity of speed demanded from the modern gas engine has led to improved methods of governing. Formerly the governor controlled the 'hit-and-miss' mechanism, and this is still the simplest arrangement where some irregularity of speed is of no consequence. Westinghouse introduced the system of controlling the quantity of mixture by throttling, and this gave excellent results, but led to bottom loops being formed on the indicator diagram and wasted power. Mr. C. E. Sargeant, of America, invented a means of cutting off the supply of mixture at varying points of the suction stroke, which gave the desired result without wasting power. This system is the correct one, and is being adopted by the leading makers one after another, each having his special design of mechanism. Fig. 70 shows the cut-off gear patented by Crossley Brothers. The richness of the mixture remains the same, while the quantity admitted to the cylinder is automatically varied by the governor for each impulse of the engine according to the power required at the moment. The cut-off valve is placed between the admission valve and the air suction and gas valve. It is a cylindrical valve, with circumferential ports sliding axially inside the casing of the admission valve, which is also fitted with corresponding circumferential ports. The cut-off valve is in equilibrium, and is opened by an ordinary eccentric and rod on a side shaft, which works a rocking lever oscillating on a pivot, the position of which pivot is determined by the governor from time to time; the action of the governor being to turn a screw with right and left-hand threads on its ends, and so to draw the pivot towards or from the motor cylinder, according to the speed of the engine. Fig. 71 shows a series of diagrams taken from an engine fitted with the cut-off valve, and the wide range of power obtainable without missing an explosion will easily be seen."

XV. *Dunlop's Patent Combined Steam and Pneumatic Marine Engine Governor*, manufactured by David J. Dunlop



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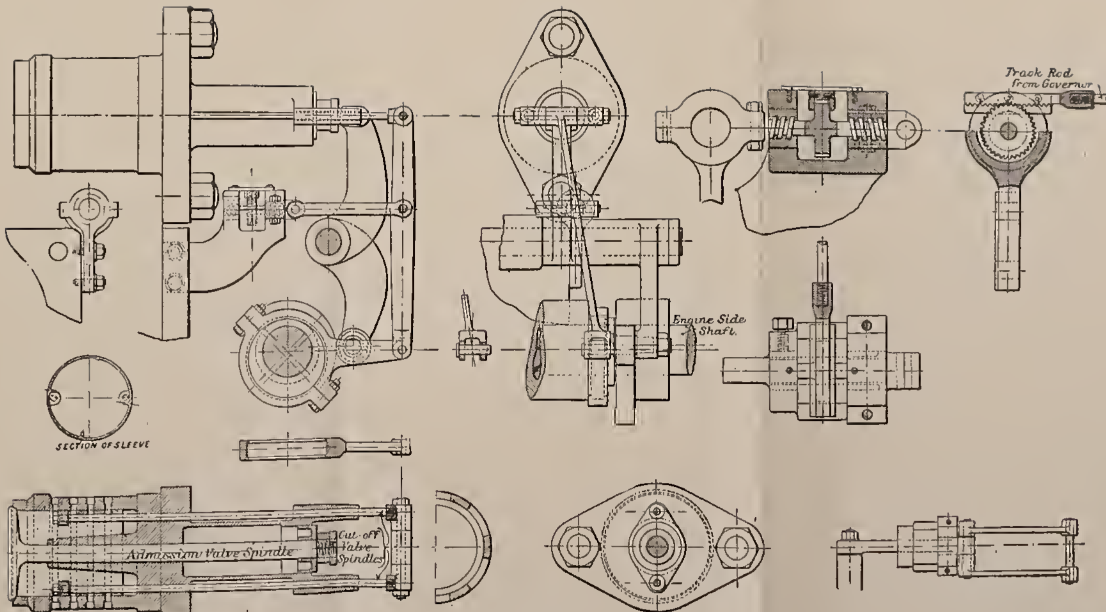

Track Rod
from Governor
Engine Side
Shaft.
SECTION OF SLEEVE
Admission Valve Spindle
Cut-off
Valve
Spindles

Fig. 70.

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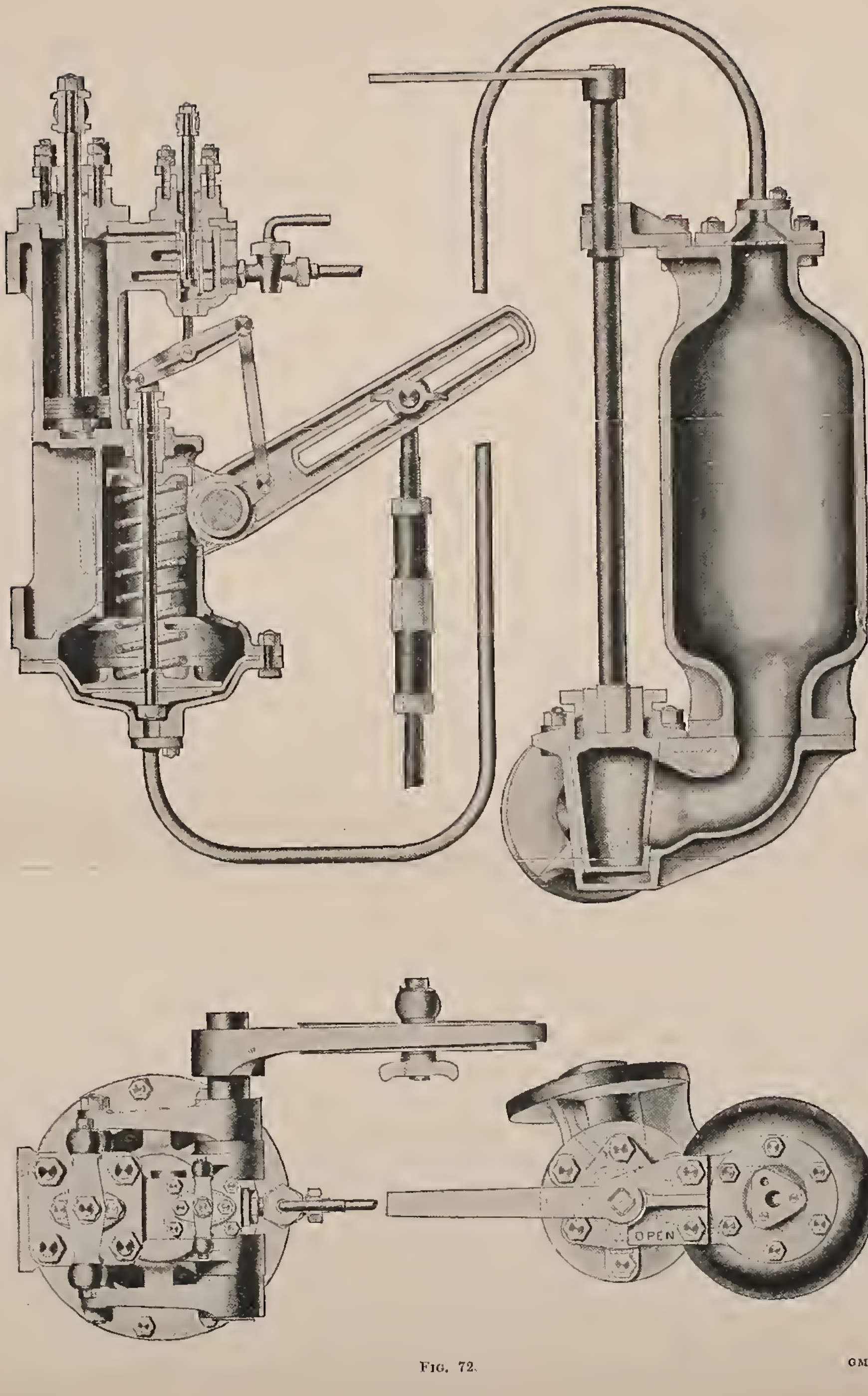

Fig. 72.

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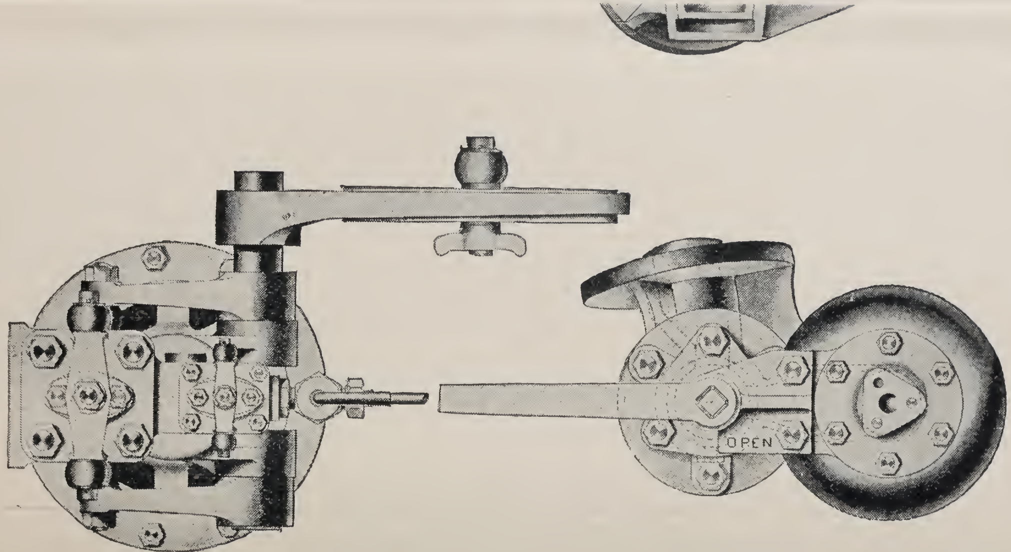

Fig. 72.

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and Co., Port Glasgow. Figs. 72, 73, and 74 show this governor.

"For regulating and controlling the speed of machinery of steam ships in heavy weather," say the makers of this governor, "it is necessary that an attendant be in charge of the steam admission valve to operate it in accordance with the varying immersion of the vessel's stern, and so prevent

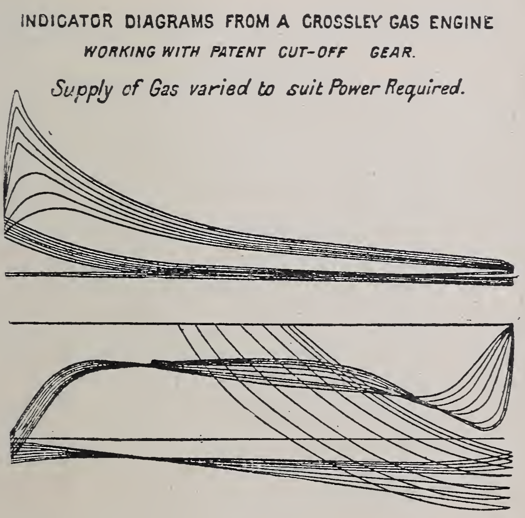


FIG. 71.

'racing' of the engines, with its consequent liability of breakdown, increased wear and tear of moving parts, and excessive waste of steam.

"If an automatic mechanical apparatus be available, which will not only take the place of the attendant (leaving him free for other duties), but at the same time be certain and regular in action, its advantages cannot be overrated, as the benefits derived from it, if only through the saving of steam, far outweigh the first cost.

"To properly control the speed of the engines, it is necessary that the steam valve shall be opened and closed slightly in advance of, and in exact accordance with, the

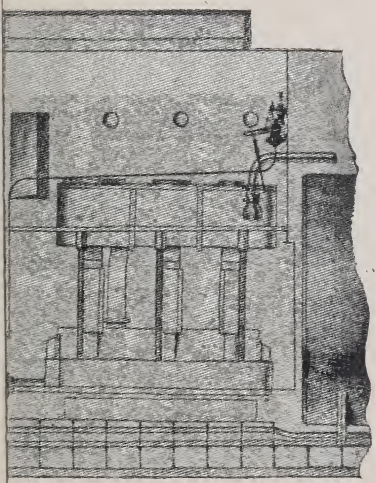
varying immersed surface of the propeller, and it is only possible to do this when the power to work the steam valve is obtained directly from the varying immersion of the propeller or head of water at the stern of the ship. This is done in Dunlop's governor.

"This governor consists of a sea-cock at the stern of the ship, opening into an air vessel or air chamber, so constructed that, by opening the sea-cock, water is allowed to flow into the air vessel, and compress the air contained therein to a pressure equivalent to the head of water outside the ship.

"From the top of the air chamber a pipe is led to the under side of an airtight elastic diaphragm, forming part of an apparatus in the engine room. On the top or upper side of the diaphragm there is a spiral spring, with means of adjusting its compression to balance the air pressure below the diaphragm. From the centre of the diaphragm a connection is made to the slide valve of a small steam cylinder, so constructed that its steam piston, which is connected by suitable gear to the steam valve of the engines whose speed is to be controlled, moves in exact accordance with the movements of the diaphragm.

"The sea-cock being open, any variation of head of water outside the ship is accompanied by an inflow or outflow of water through it, and consequently a variation in the pressure of the air contained in the air vessel and under the diaphragm of the engine-room apparatus, causing the diaphragm to move through such a part of its travel as is requisite to enable the compression of spring and air pressure to balance one another again. Every movement of the diaphragm is followed by a corresponding movement of the governor steam piston, and consequently of the steam valve of the engines under control. The time required between the variation in the head of water at the stern of the ship and the moving of the steam valve being, so far as can be judged from careful experiments, absolutely *nil*.

"The most important feature of Dunlop's governor is the fact that it will anticipate any increase in the speed of the engines due to the propeller rising out of the water—a feature which can be claimed for no other governor, as they



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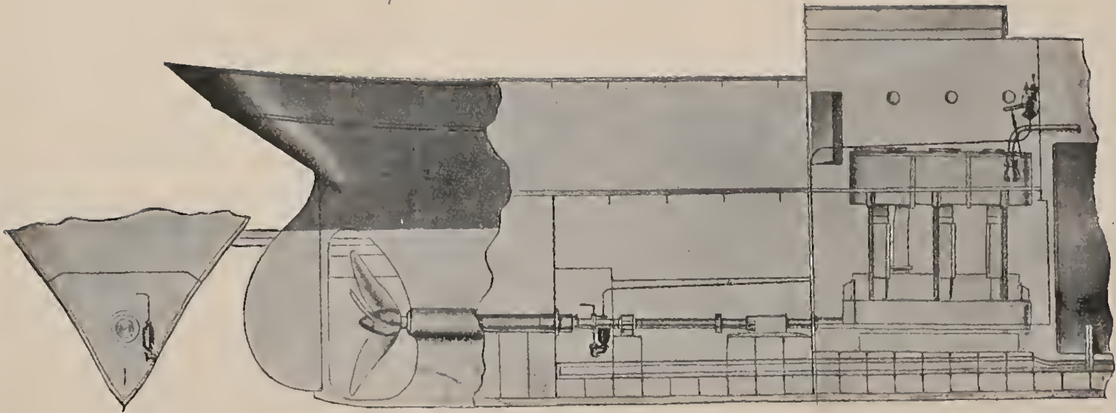

Fig. 73.

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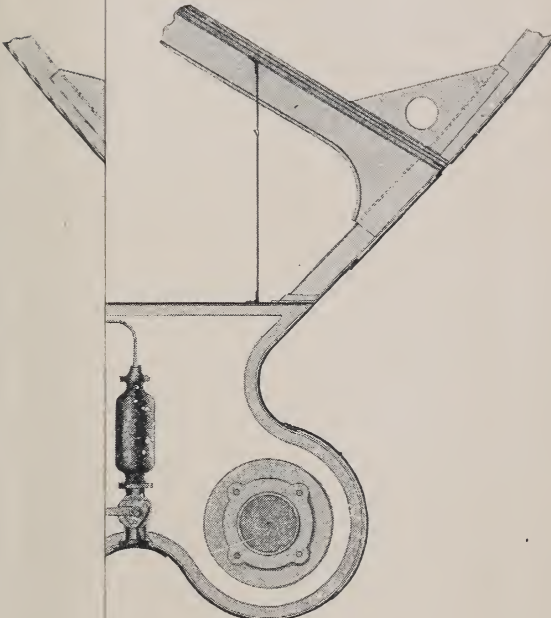

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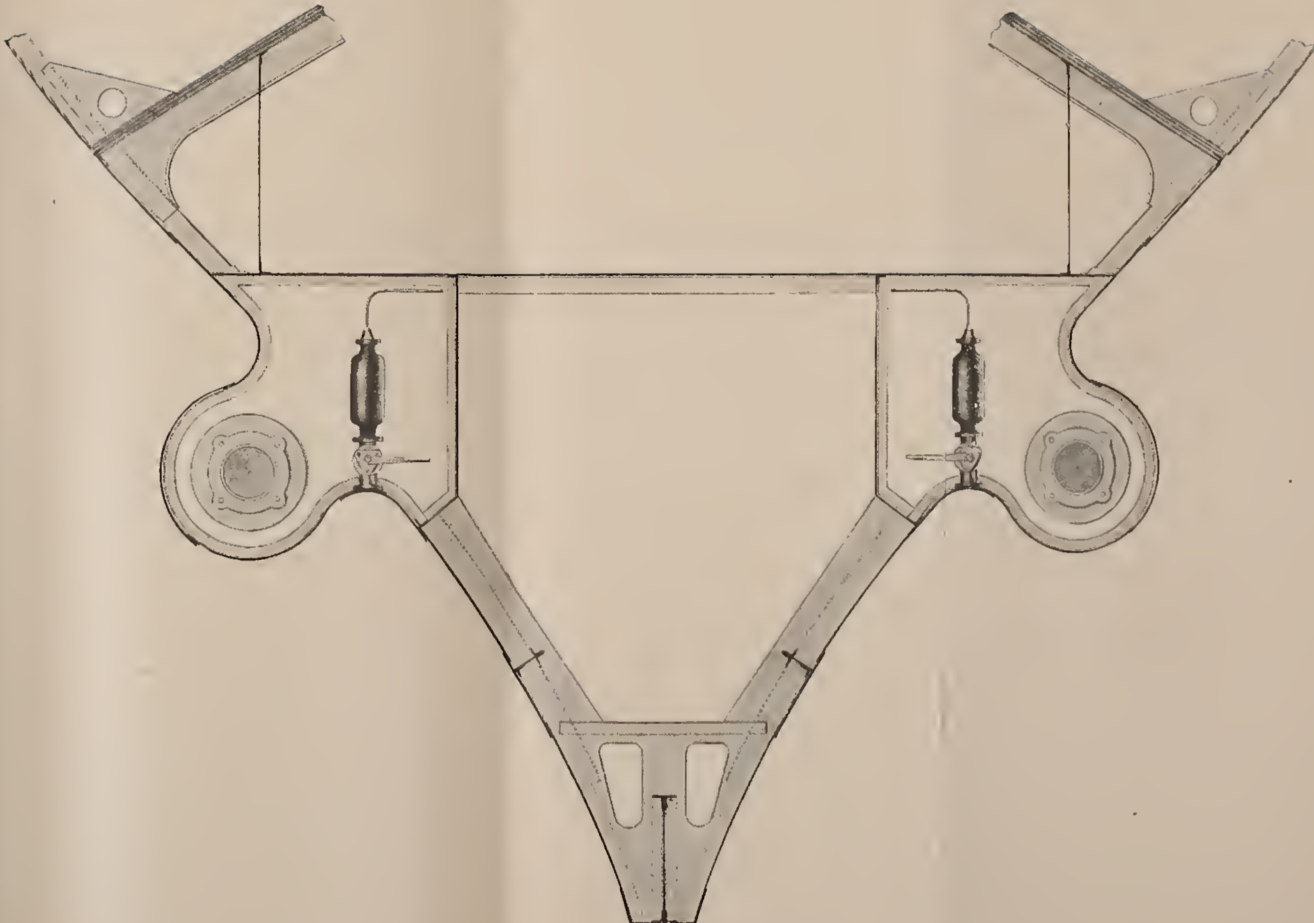


Fig. 74.

GM

one and all depend upon a variation in the speed of the engines to be controlled before the power to act is available. This anticipating action of the governor is obtained by adjusting the balance between the spring and the air pressure under the diaphragm, so that the diaphragm begins to fall and the steam valve to close when the tips of the propeller blades are anything from one to several feet under the surface of the water.

"With the multiple cylinder expansion engines now general, unless a governor has this anticipating power it is practically of no use, the quantity of steam contained in the casings and pipes being sufficient of itself, when expanding into the condenser, to cause the most serious 'racing.'"

XVI. *Aspinall's Patent Marine Engine Governor*, manufactured by Aspinall's Patent Governor Company, Liverpool. The principle on which this governor acts is the same as that of the gas engine inertia governor.

The governor consists of a loose weight carried in a frame which is bolted to the air pump lever or to a special reciprocating lever fitted to the engine for the purpose. A lever, connecting to the main steam or throttle valve, is carried on a pin concentric with the shaft on which the air pump lever works, in a separate bracket close in front of the air pump lever carrying the governor. To prevent the loose weight being left behind the air pump lever as it descends when working at its normal speed, it is held in place by a small spring buffer, but should the speed of the engine suddenly increase through the propeller being lifted too far out of the water, the inertia of this weight causes it to overcome the resistance of the buffer and so lag behind, at the same time operating a pawl, which shoots out and engages with the under side of the throttle valve lever before mentioned, and this, on the next up stroke of the air pump lever, is carried up with it and shuts off the steam supply.

The weight, on resuming its normal position due to the slowing down of the engine, shoots out a pawl, which engages with the top side of the throttle valve lever, also, at the same time, withdrawing the one on the lower side, and thus brings the lever back and again opens the throttle valve.

The spring buffer, which prevents the weight being left behind when the engine is running at its normal speed, can be adjusted as required.

In addition to the above, this governor is fitted with an "emergency gear," which consists of an extra small weight, which on coming into action, when an excessive race occurs, operates a latch which locks the pawls and weight in their shut off position and entirely prevents the re-opening of the throttle valve.

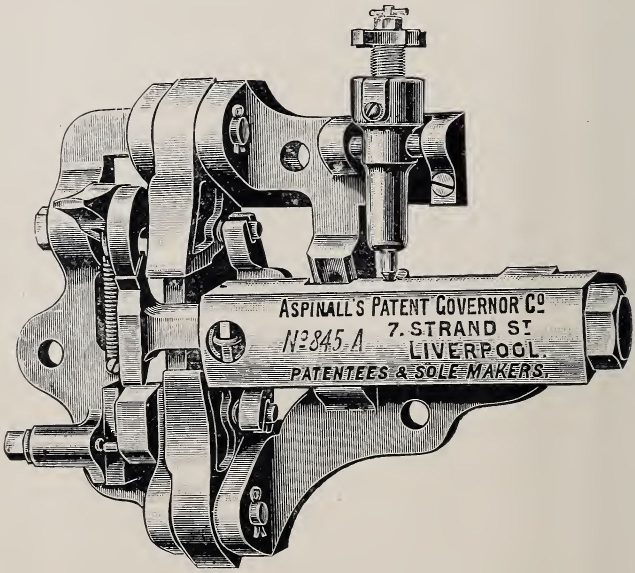
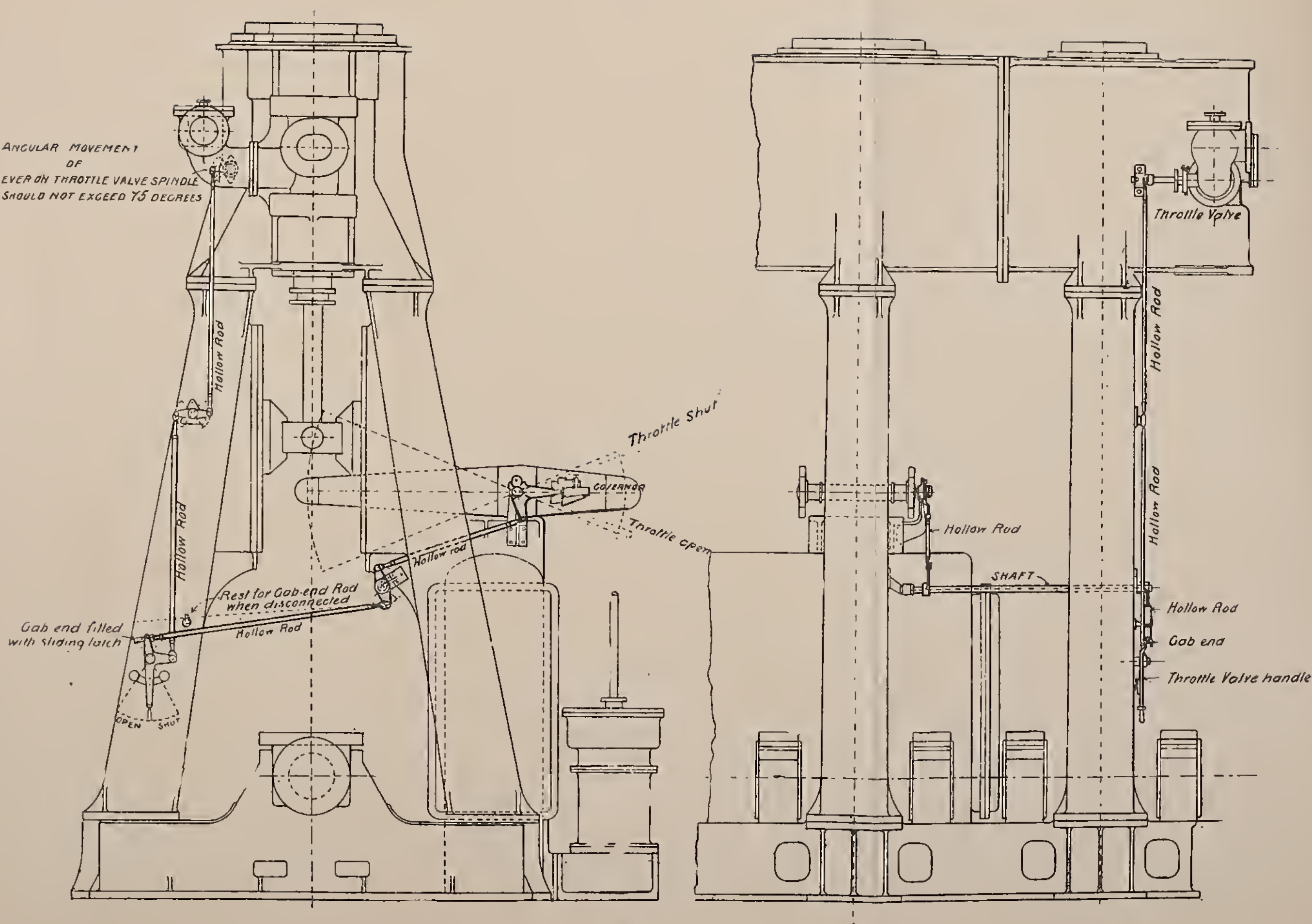


FIG. 75.

This governor is not only used on board ship, but has recently been introduced as an emergency governor into several central power stations to prevent accidents from racing when the governor proper has been put out of action through the failure of some part of the mechanism between engine and governor.

Fig. 75 is an external view of this governor.

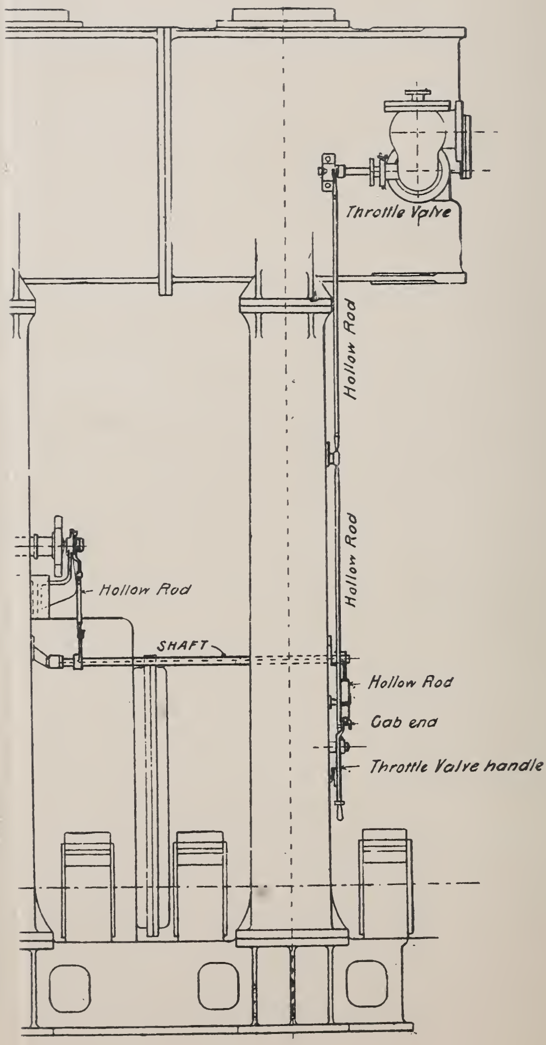
Fig. 76 shows the governor in position on pump end of lever.



ANGULAR MOVEMENT
OF
LEVER ON THROTTLE VALVE SPINDLE
SHOULD NOT EXCEED 75 DEGREES
Hollow Rod
Hollow Rod
Throttle Shut
GOVERNOR
Throttle open
Hollow rod
Rest for Gab-end Rod
when disconnected
Gab end fitted
with sliding latch
Hollow Rod
OPEN
SHUT
Throttle Valve
Hollow Rod
Hollow Rod
Hollow Rod
SHAFT
Hollow Rod
Gab end
Throttle Valve handle

FIG. 77.

GM


Throttle Valve
Hollow Rod
Hollow Rod
Hollow Rod
SHAFT
Hollow Rod
Cab end
Throttle Valve handle

GM

Fig. 77 shows the general arrangement of governor on the engine.

The following is a description by the makers of this governor:—

"The governor consists of a hinged weight W, operating two pawls, P P, carried on a frame which is bolted to air

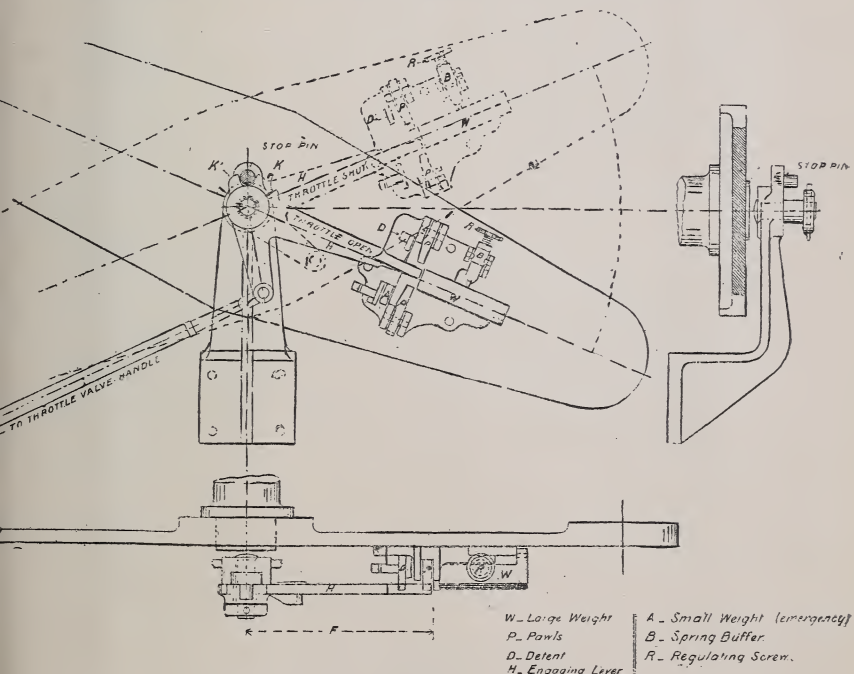


Fig. 76.

pump lever or other reciprocating part. When the revolutions of engines are increased by about 5 per cent above the normal speed the weight W is left behind, and reverses the position of the pawls, causing bottom one to engage with lever H (fig. 76), lifting it throughout the whole upward stroke, and thus shutting off steam by closing throttle valve. On the return stroke the detent D is lifted, liberating

weight W, and, if the revolutions have moderated, the position of the two pawls is again altered, the top pawl now engaging with lever H, depressing it, and thus re-opening the throttle valve. The emergency gear only comes into operation in case of a very excessive race, such as losing a propeller or a broken shaft, in which case the small weight A is left behind and locks the weight W in the 'shut-off' position, thus effectually preventing the re-opening of throttle valve. To unlock the emergency gear press the large weight W upwards from the under side, when the small weight A will fall out of gear."

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John Heywood, Excelsior Printing and Bookbinding Works, Manchester.